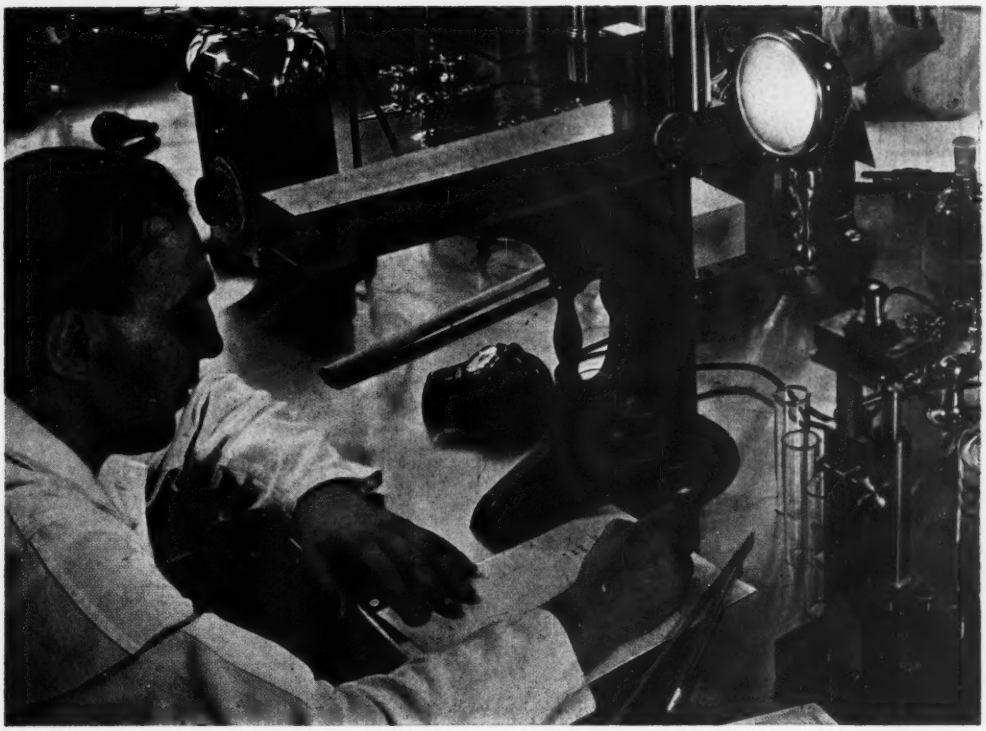
Vol. 13, No. 1 (Part 1) January, 1957

# LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE



**Laboratory Testing of Petroleum Products**

(Cover Story, p. 11)

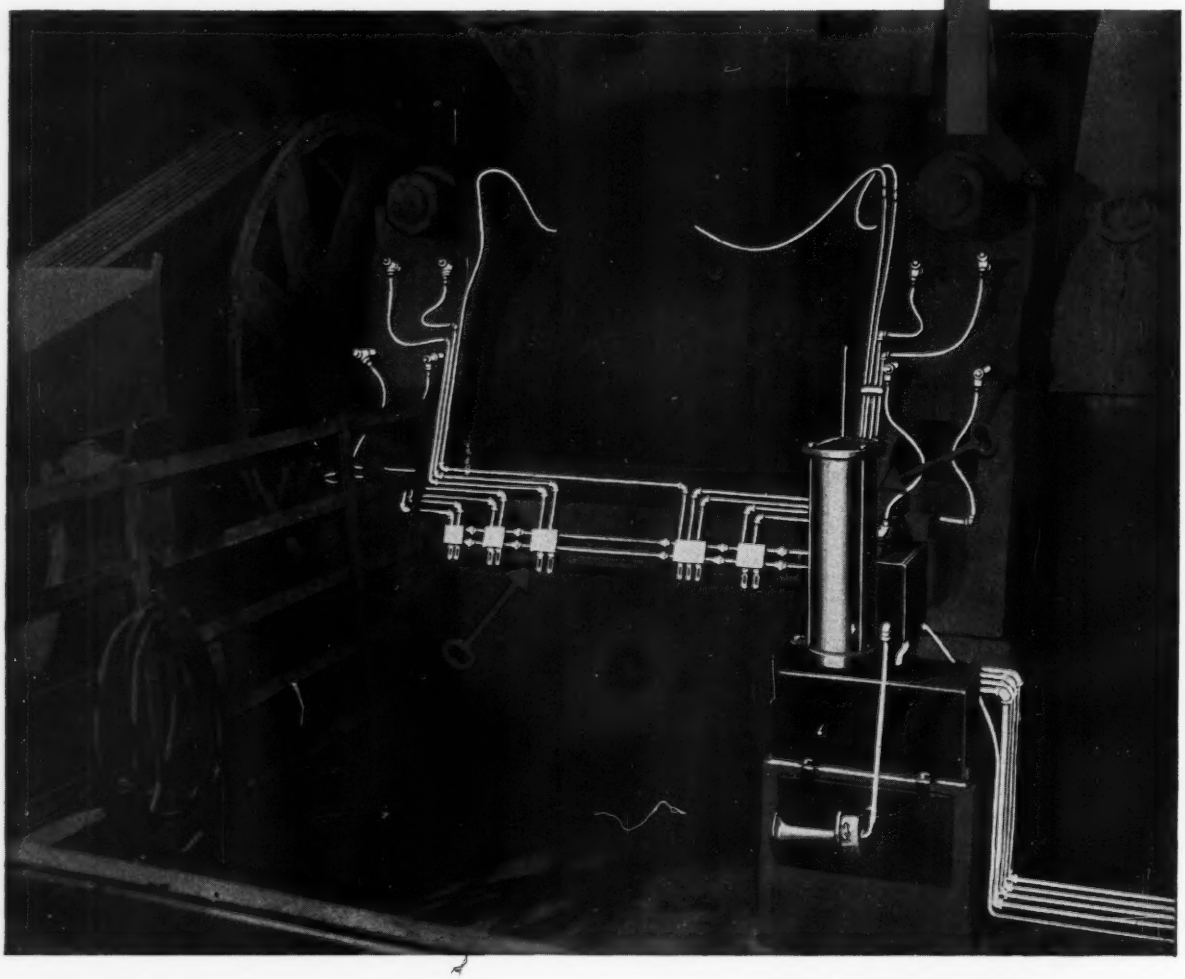
**STARTING THIS ISSUE:**

*Symposium On Industrial Dermatoses*

JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

# Crusher bearings lasted 13 months, but with Farval no new ones in 6½ years

FARVAL—
*Studies in Centralized Lubrication*
No. 188



● Back in 1949, an automatic system of Farval Centralized Lubrication was installed on this Fairmount Crusher. It paid for itself in 4 months by the savings in lubricant and labor that resulted.

In the nearly 7 years of its operation, Farval has returned rich dividends. Before the system was installed the life of a set of bearings was only 13 months. Today the identical bearings installed when the lubrication system was applied are still in service. Maintenance cost has been negligible. This success record led the cement company to specify Farval for a new mill crane and a power shovel.

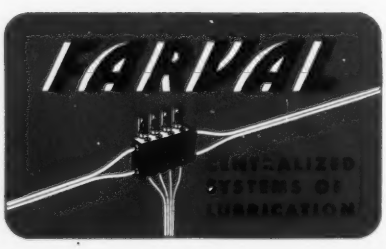
Investigate Farval for your equipment. Write for Bulletin 26-R. The Farval Corporation, 3267 East 80th Street, Cleveland 4, Ohio.

*Affiliate of The Cleveland Worm & Gear Company, Industrial Worm Gearing. In Canada: Peacock Brothers Limited.*

**KEYS TO ADEQUATE LUBRICATION—**

**Wherever you see the sign of Farval—the familiar central pumping station, dual lubricant lines and valve manifolds—you know a machine is being properly lubricated.**

**On this crusher in an eastern cement plant, savings in lubricant and labor, due to Farval, run well over $375 a month.**



# LUBRICATION ENGINEERING

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Volume 13, No. 1

January, 1957


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# The Sight Glass

## Highlights of Articles Scheduled for Coming Issues!

**Chemical Composition of Lubricating Oils, Cutting Oils, and Petroleum Solvents**

To pinpoint some of the agents often suspected as causing dermatosis within a plant, various lubricating oils, cutting fluids, and petroleum solvents are discussed in the light of their chemical compositions.

**Skin Diseases Caused by Cutting Oils**

Contact with cutting oils may cause various diseases of the skin among which are: oil acne, dermatitis, infected wounds, and less occasionally, granuloma, melanoderma, chlor-acne, and cancer. Some of the characteristics of the diseases are treated so that supervisory personnel may recognize the symptoms should they occur and refer the employee to proper medical care.

**Predicting Performance of Starved Bearings**

Several types of journal bearings operate under conditions properly called starved lubrication. In certain cases, such as wick-fed and oil-ring bearings, the rate of oil feed must be determined empirically by experiment. When the rate of oil feed to the bearing is less than the normal end leakage, which is caused by the load-carrying pressures generated in the oil film and by the oil-feed pressures, starved lubrication occurs. Speeds and loads which are too high may often cause this condition. A design procedure for a starved bearing is outlined. By extending the analysis, the running time of the bearing after the oil feed has been cut off can be determined.

**Numerical Solution of Reynolds' Equation for Sector Thrust Bearings**

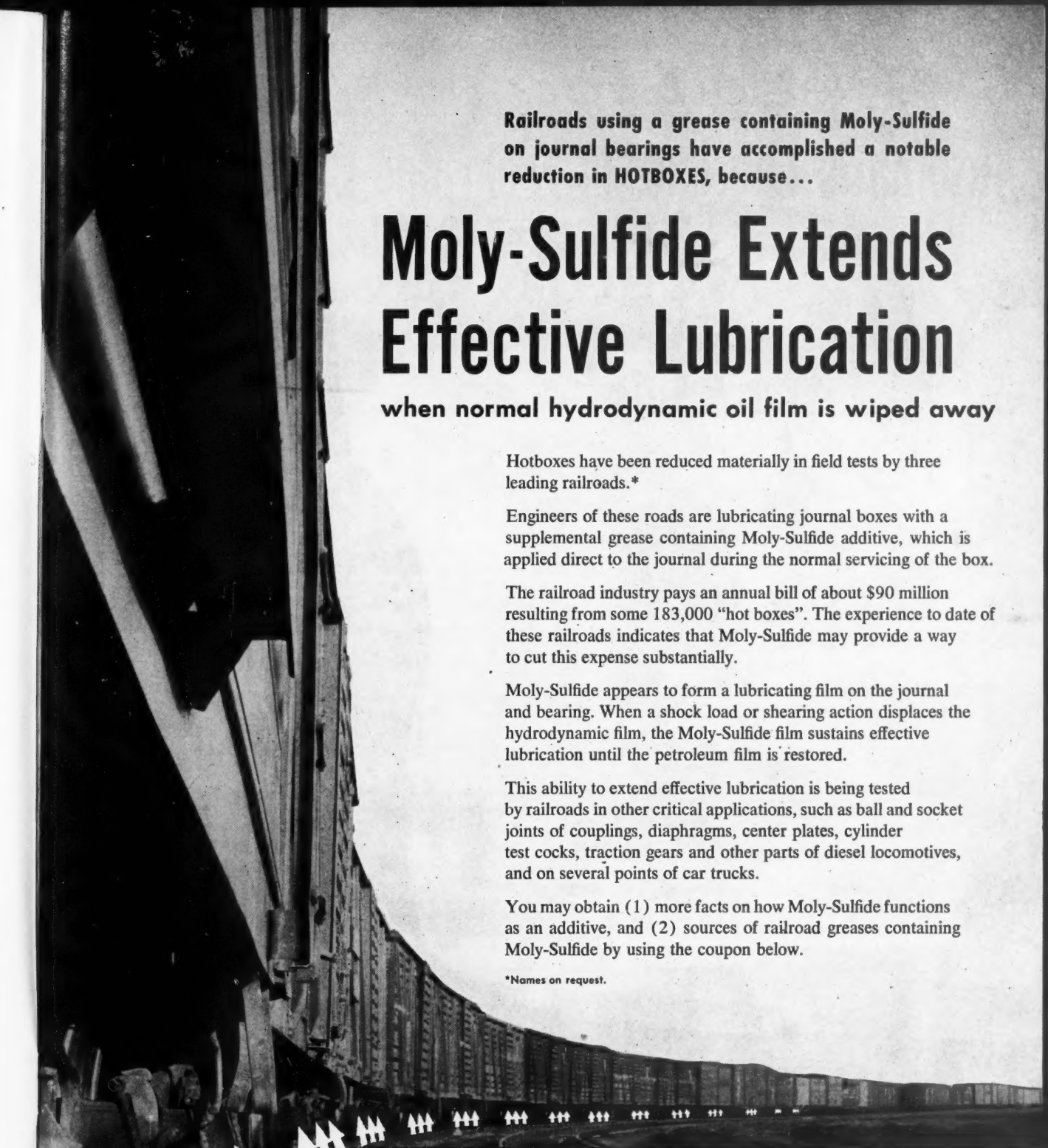
A method of solving the Reynolds' equation using digital computer techniques has been developed to predict sector bearing performance. The results compare favorably with experiment and known analytical solutions. The investigation indicates that good design criteria may be established for sector thrust bearings with arbitrary geometries, and possibly very accurate predictions may be obtained if the Reynolds' and energy equations can be solved simultaneously by this technique.

**Lubrication of Friction Drives**

Friction drives are frequently employed when continuous variation in the speed ratio of coupled shafts is required. Power transmitting elements are heavily loaded to obtain the high frictional forces necessary and lubricants possessing a high coefficient of friction are desirable. A test apparatus is described which involves two rotating balls pressed together so that the frictional behavior can be measured in the region of speeds about the nominal rolling condition. The role played by the temperature coefficient of viscosity of the lubricant in determining the efficiency of the frictional drive is discussed.

**Studies of Formation and Behavior of an Extreme-Pressure Film**

Gears lubricated with a mineral oil compounded with dibenzyl disulfide containing radioactive sulfur, $S^{35}$, were operated under controlled conditions. The formation and behavior of the resultant sulfur-containing film on the load-bearing surfaces have been studied, using the beta particles from the sulfur as a means of detection of the film and measurement of its thickness.



**Railroads using a grease containing Moly-Sulfide on journal bearings have accomplished a notable reduction in HOTBOXES, because...**

# Moly-Sulfide Extends Effective Lubrication

## when normal hydrodynamic oil film is wiped away

Hotboxes have been reduced materially in field tests by three leading railroads.*

Engineers of these roads are lubricating journal boxes with a supplemental grease containing Moly-Sulfide additive, which is applied direct to the journal during the normal servicing of the box.

The railroad industry pays an annual bill of about $90 million resulting from some 183,000 "hot boxes". The experience to date of these railroads indicates that Moly-Sulfide may provide a way to cut this expense substantially.

Moly-Sulfide appears to form a lubricating film on the journal and bearing. When a shock load or shearing action displaces the hydrodynamic film, the Moly-Sulfide film sustains effective lubrication until the petroleum film is restored.

This ability to extend effective lubrication is being tested by railroads in other critical applications, such as ball and socket joints of couplings, diaphragms, center plates, cylinder test cocks, traction gears and other parts of diesel locomotives, and on several points of car trucks.

You may obtain (1) more facts on how Moly-Sulfide functions as an additive, and (2) sources of railroad greases containing Moly-Sulfide by using the coupon below.

*Names on request.

*Use the moly key ...to lubrication insurance*

- Films on metal
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Resists:

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Literature

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# Design for

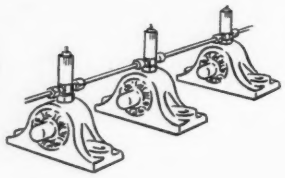
# Automatic Lubrication From One Central Point

## with an ALEMITE Accumatic® System

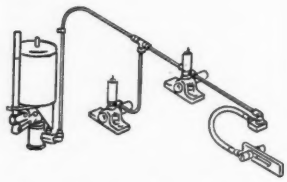
***FASTER . . . FOOLPROOF . . . LESS COSTLY!***



Replace grease cups or grease fittings with Accumatic fittings. (Available in same thread sizes as grease fittings, cups.)



Connect Accumatic fittings with copper tubing. (Alemite has tubing, clips and accessories for easy installation.)



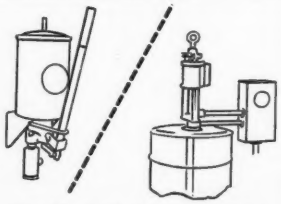
Connect sliding, rotary or oscillating parts into tubing system. (Flexible hose and swivels for moving parts.)

**Type 1 Accumatic Valves**

For fluid oil or light grease. In range of sizes, delivering from .005 to .100 cu. in. of lubricant. Various shapes: Tees, straight-thru, inserts, angles. Spring pressure provides gradual feed. Adjustable or fixed output. System serves up to 400 bearings. Manual or automatic operation available.

**Factory-tested—field-proved!**

Grueling field tests show no appreciable variation in the amount of lubricant discharged after 73,312 lubrication cycles—equal to *122 years of twice-a-day service!*



Provide central pump to supply lubricant to system. (Ordinary hand pump or fully automatic barrel pump.)

### ALEMITE ACCUMATIC ADVANTAGES!

- Prevents application of wrong lubricant.
- Seals completely against damaging dirt, grit and water.
- No parts are neglected—lubricates inaccessible and dangerous bearings at regular intervals.
- Eliminates product spoilage due to over-lubrication.
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- Delivers *exact* amount of lubricant to bearing.

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# LUBRICATION ENGINEERING

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## Where We Stand

*"The American Society of Lubrication Engineers dedicates its Journal to increasing the fund of knowledge in the field of lubrication; to making available the discoveries of research to applied lubrication; to giving recognition to the importance of lubrication and the Lubrication Engineer in Industry; to fostering research, data, techniques, and news as an impartial organ of lubrication."*

Lubrication Engineering, June, 1945.

*" . . . lubrication is . . . crossing the threshold of a new era of intense and important scientific development which is rich in the promise of better things to come. The availability of a new medium wherein new knowledge, ideas, experiences, and suggestions can be assembled, correlated, weighed, and suitably presented will play a key role in the advancement of this infant science. The ASLE has as one of its leading goals the fulfillment of this need through its official publication, Lubrication Engineering."*

E. M. Kipp, June, 1945.

In the eleven years since the founding of Lubrication Engineering, many ideas and concepts of lubrication and lubricants have changed. The infant science has grown to a veritable giant encompassing the fields of chemistry, physics, mechanics, and metallurgy. Research in lubrication has grown correspondingly, such that publication of valuable data now lags the actual work performed by as much as two years.

Growth of the journal can be measured in many terms. Total published pages during 1945 totaled 140. Last year 448 pages were published of which more than 167 dealt with technical matter and 133 with news, new products, technical literature, and abstracts of technical information. Since its founding, Lubrication Engineering has published 309 technical papers totaling 1373 pages plus an additional 1190 pages of miscellaneous information. Most significant, however, is the addition of several new features of topical interest and the plans for new features to be added during 1957.

Machine tools, labor, and performance standards have improved our standards of living. Each improvement is the result of better understanding the role of lubrication to increase the machine efficiencies of modern production equipment. The need for "productive maintenance" programs within the plants has lead to greater acceptance by management of the lubrication engineer as the solution to reduced costs in spite of more costly labor, equipment, and raw materials. Lubrication Engineers have become modern pioneers exploiting the last remaining production frontier — improved methods.

Automation has had its impact on the lubrication engineering function in the modern mine, mill, and factory. Without informed lubrication engineers we could not have achieved the current high levels of production efficiency — would not benefit from continuous nor even continual production.

The function of this journal in the great period of change throughout the past 10 years has never deviated. It will continue to correlate, weigh, assemble and present all aspects of this fundamental science — to offer a sounding board for new ideas, concepts, and methods.

Before you is the first monthly issue of Lubrication Engineering. It represents attainment of one goal of the founders of ASLE. With humility, the editors and staff of Lubrication Engineering reaffirm their pledge to provide the most thorough and impartial coverage of the science of lubrication in the years to come — to provide greater understanding and mutual appreciation for the so-called "longhair" article and the "practical" article, which *together* provide the very foundation of this science.

R. D. McCormick
ASLE Publications Manager

# MOLYKOTE®

## CHECKS "STICK-SLIP" ON A GRAND SCALE

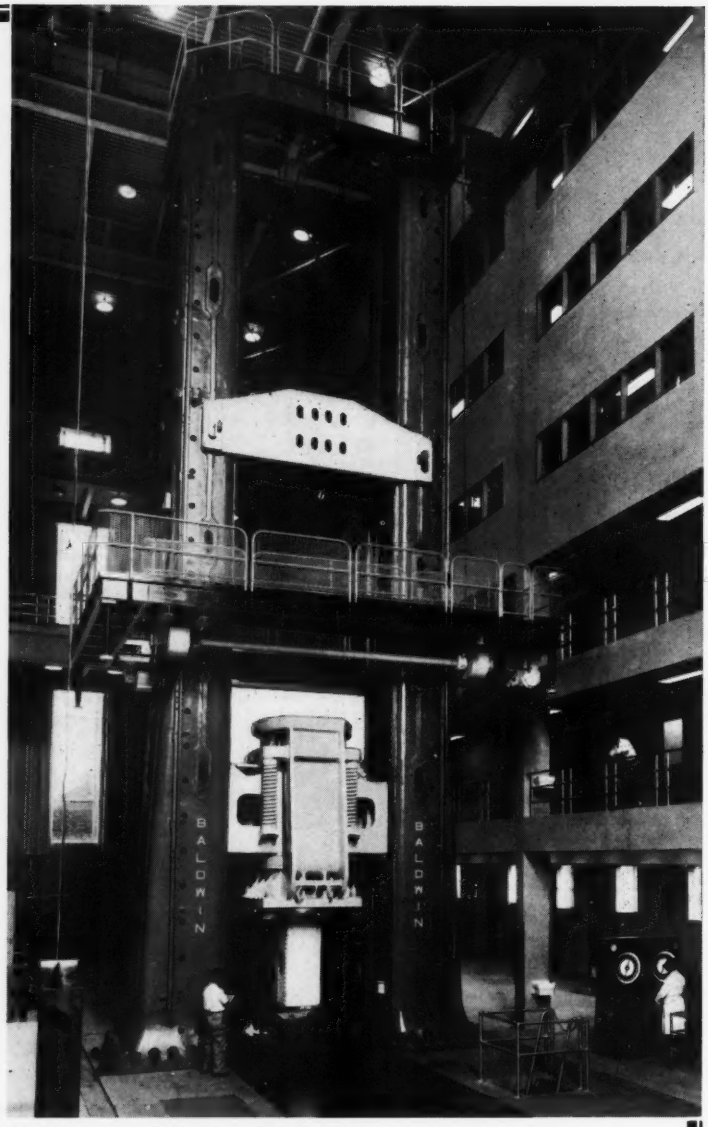
When engineers of Baldwin-Lima-Hamilton Corporation designed and built this mammoth 5,000,000 pound universal testing machine for Lehigh University, they realized from years of experience that extreme testing pressures could cause destructive "stick-slip" action in the test specimen grips and the loading screws.

That is why MOLYKOTE Lubricant was generously applied to both the grips and the screws before the first trial specimen was broken in this giant universal tester.

Baldwin engineers, as well as engineers throughout industry, have come to depend upon MOLYKOTE Lubricant in extreme pressure applications.

MOLYKOTE is manufactured from the purest molybdenum disulfide powder available to industry today. It is compounded under laboratory supervision to retain this purity in the manufacturing process and has achieved an enviable reputation for overcoming the toughest lubrication problems in industry.

MOLYKOTE Lubricant is distributed throughout the world, wherever industry faces the obstacles of providing adequate lubrication in extreme pressure and extreme temperature applications.




5,000,000 pound B-T-E Universal Testing Machine installed in Fritz Engineering Laboratory, Lehigh University, Bethlehem, Pa. Built by Baldwin-Lima-Hamilton Corporation, Eddystone, Pa.

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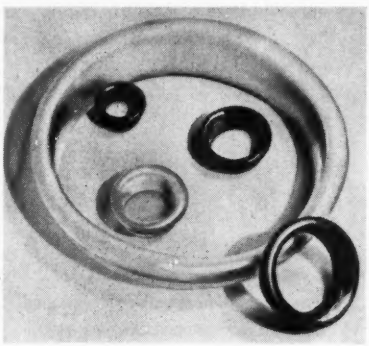
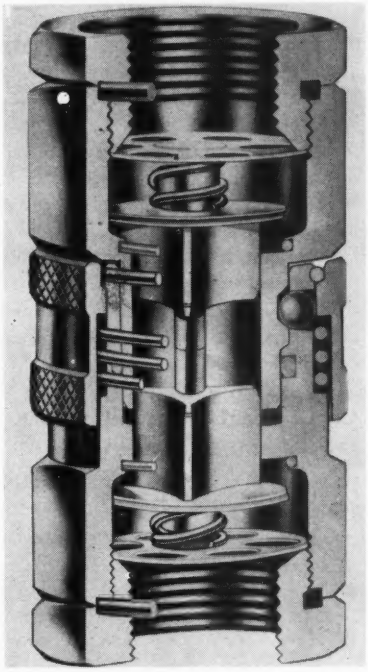
# New Products

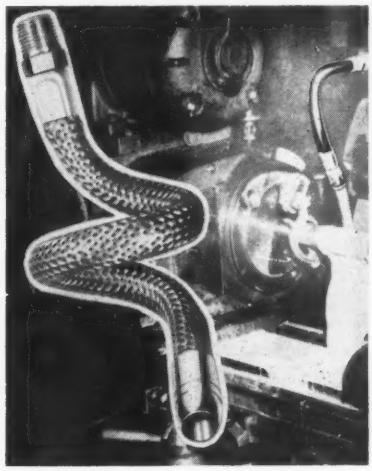
**Industrial Cooling & Lubricating System.** Liquidaire, a new industrial cooling and lubricating system that cuts metal machining time up to 75%, has been developed especially for tool room and single-point metalcutting and stamping operations. The ruggedly constructed system operates on the principle of mist cooling. A special Liquidaire coolant is pressurized and forced through the nozzle; as the fluid is ejected, it flows over the outside of the nozzle's liquid control needle. Through the center of this needle, a jet of high-velocity air is discharged, penetrating the liquid from the inside. This action produces liquid suspension in the air which is controlled by a patented ejector. Upon expansion in the atmosphere, mechanical refrigeration averaging 25 degrees lower than compressed air temperature is directed on the cutting tool and material being worked. As the pressurized mist coolant strikes the cutting tool, the heat caused by the tooling operation is transferred to the coolant vapor and evaporated into the atmosphere. The lubricating properties of the coolant remain on the tool, forming a protecting film over the cutting surfaces. (Frisch Corp., 1400 W. Wabansia, Chicago 22, Ill.)



**Rolled Washers.** "Nyltite Rolled Washers" (pictured above) — a revolutionary new fastening improvement made from duPont's Zytel nylon — have been introduced by the Nyltite Corp. of America. The rolled washers are slipped over the bolt, screw, rivet, or nail and, when pressure is applied to tighten the fastener, "cold-flow" around the fastening device and under the fastener head. The product's unique hugging qualities fill all space between threads and hole providing perfect air-tight, leak-proof seals. The new washers also act as a locking device, prevent galling, and dampen the effect of vibration. (Nyltite Corp. of America, Union, N. J.)



**Quick Connect-Disconnect Coupling.** A new quick connect-disconnect coupling (pictured above) for acids, alkali, solvents, high-pressure steam . . . for previously impossible fluid transfer . . . has been developed that will function perfectly through a —100° to +500° F. temperature range. Known as the "HK" coupling, its use is limited only by the ability of its Teflon nipple seals and valve seals — and there is no known solvent for Teflon. Manufactured from solid bar stock, the coupling can be furnished in 303 and 316 stainless steels, brass, aluminum, special carbon steel, or from any machinable material. It is available plain (without valves) or with automatic shut-off valves incorporated in either or both ends (causing the valved end of the line to shut off immediately and automatically upon disconnection; when connected, the coupling valves automatically open). The coupling is manufactured in ½" through 3" I.D. pipe sizes with any type end connection desired. (Snap-Tite, Inc., Union City, Pa.)



**Plastic Coolant Hose.** A new coolant assembly (pictured above) that practically makes metalworking a white-collar job has been accomplished by a unique plastic coolant hose that can be twisted and turned into any bend and stays set in that position. Appropriately named the Kling-Flo, its one-piece aerator nozzle eliminates all splash and coolant throw-off, and vigorously mixes millions of tiny oxygenized air bubbles into the water-soluble coolants used on most metalworking machinery. These bubbles are formed just as the liquid coolant leaves the assembly; the bubbles never dissipate or leave any residue or scum on the work. A special steel-clad armor protects the flexible plastic coolant hose from the usual hot, sharp spiral cuttings and chips thrown off the work. Its flexibility allows the assembly to assume any bending

(Continued on p. 56)

# ASLE News Notebook

Compiled by
R. D. McCormick

## ASLE Personalities

**W. P. Youngclaus, Jr., admires the ASLE tie clasp worn by Mr. K. Sitharama Rao, Lubrication Engineer from Madaras, India, during his recent visit to the National Office.**

The specifying power of the technical man in India is far greater than that observed for the engineer in the United States according to Mr. K. Sitharama Rao, Engineer, Technical Department, of the Standard Vacuum Oil Company, Madaras, India. Mr. Rao was a recent visitor to the National Office during his inspection tour of US Industry.

Labor in India, being much less expensive, is not a deciding factor in problems such as reclamation of costly lubricating oils and cutting fluids. Because of his ability to salvage such valuable materials the lubrication engineer is a highly respected man in India.

Mr. Rao was alarmed at some of the practices he observed in plants in this country which use cutting fluids. In India cutting fluids are not used quite as recklessly. He feels that the man in the plant should be made more aware of the function of protective creams and ointments in preventing any potential health hazard. (Editor's Note: Lubrication Engineering is currently featuring a series of articles on Industrial Dermatosis.)

Among other major differences in the problems confronting the Indian lubrication engineer and his American counterpart is that of locating readily available sources of supply for dispensing and handling equipment. Mr. Rao hoped that Lubrication Engineering would be able to publish more information on lubrication equipment in future articles.

ASLE publications for use in the field were praised by Mr. Rao. In one instance he was called in on an apparent bearing failure involving rolling contact bearings. By comparing the bearings in question with photos in the ASLE publication "Interpreting Service Damage in Rolling Type Bearings" he was able to determine that no lubricant failure had occurred, but that shock scoring was present. Tracing the shipment of other bearings Mr. Rao found that repeated shock was encountered during rail transit because of poor packaging. This scoring led to the failure encountered. All future shipments were made by air to prevent recurrence of the shock condition.

## Committee Reports

### Properties of Lubricants

At a meeting of the committee in Atlantic City, October 9, 1956, the members of the committee agreed to sponsor one session at the April Annual meeting of ASLE. The committee recommended to the Technical Advisory Board that the Society prepare a comment or recommendation on the proposed viscosity classification system to be presented during a symposium to be held at the 1957 meeting of the ASTM in June. This system will be a result of several proposals currently being screened by sub-committee U-III-B of the ASTM. The research sub-committee of this committee is defining areas of work which should be supported by the ASLE.

### Seals and Packings

Four papers were discussed for sponsorship by the committee for the 1957 Annual Meeting. A series of articles describing the basic description, application, and limitations of various types of seals was suggested as an educational feature for Lubrication Engineering. The committee recommended that various manufacturers of seals and packings be contacted to supply these articles for editorial consideration. The SAE Aircraft Committee has expressed interest in cooperating with the committee on problems involving seals and packings. One of the first steps to be undertaken will be a cooperative effort to establish seal nomenclature for all types of seals.

### National Railroad Lubrication Council

During a meeting of the NRLC in Chicago, September 11, Mr. E. E. Smith of the Climax Molybdenum Company reported on a research proposal to be conducted by the Armour Research Foundation to cover the use of Moylbdenum Disulfide Grease for Railroad Application. The program will investigate the application of a calcium soap grease which contains an oil of 78 SUS viscosity at 100° F and 12% Molysulfide Technical, as a supplementary lubricant for use in railroad freight car journal boxes. The investigation will include the possible effect of Molygrease upon the boundary friction properties, and wear properties of the babbitt as well as the magnitude of the friction coefficient.

After reviewing the offer of ASLE to the NRLC to actively participate as a member group of the ASLE while maintaining their present name, organization, By-Laws, and information liaison the committee accepted the following resolution:

> **"That NRLC become affiliated with ASLE as outlined in their letter of May 15, 1956 and that liaison members be notified of this proposed affiliation and asked for their sponsors opinions."**

Certain changes in the NRLC By-Laws will be necessary to permit this affiliation. The secretary was instructed to draw up the necessary changes to permit this affiliation, if the affiliation is accepted by the sponsoring societies and organizations. The changes will

be voted upon at the next NRLC meeting, to be held during the SAE meeting in Detroit, Mich., during the week of January 14-18. 1957.

### Eastern Regional Area Representatives Named

Mr. John P. Critchlow of the Gulf Oil Corporation, Pittsburgh, has been named area representative for the Pittsburgh and Wheeling, West Va. Sections. Mr. D. F. Wilcock, General Electric Co., Schenectady, N. Y. has been appointed area representative for the Buffalo, Schenectady, and Syracuse Sections; and Mr. H. A. Wilson, Commercial Filters Corp., Lexington, Mass., will be area representative for the Boston and Connecticut Sections. Area representatives assist the regional vice-president with personal liaison and Section coordination of educational courses and membership promotion.

### Philadelphia Section to Sponsor Lubrication Engineering Course

A course in practical lubrication will be sponsored by the Philadelphia Section on February 8, 1957 at the Esso District Office, Esso Road & City Line Avenue, Bala-Cynwyd, Pa., from 9:00 AM to 5:00 PM. Course fees are $10.00 for ASLE members and $15.00 for interested non-members. Topics to be covered and speakers include: "Significance of Functional Tests in Relation to Application of Lubricants in Practical Service" by C. A. Bailey, Chief Lubrication Engineer, National Tube Division, U. S. Steel Corp., Pittsburgh, Pa. "Setting up a Standardization and Simplification Program for Lubricants in Industrial Plants" by R. Kageff, Ford Motor Co., Dearborn, Mich. "Selection and Application of Lubricants for Plain Bearings" by P. H. Dreissigacker, Farrel-Birmingham Company, Inc., Ansonia, Conn. "Ball and Roller Bearing Application and Maintenance" by W. T. Everitt, Eastman Kodak Co., Rochester, N. Y. "Lubrication of Various Types of Gears and the Solution of Gear Problems in Industrial Applications" by H. O. Kron, Chief Engineer, Philadelphia Gear Works, Philadelphia, Pa. and "How to Care for Hydraulic and Circulating System Oils" by J. R. McCoy, Commercial Filter Corp., Lebanon, Indiana.

### Lubrication Engineering Course to be Held at York, Pa.

Soon after the course in practical lubrication at Bala-Cynwyd the Philadelphia Section will sponsor a one day course February 28 at the Yorktown Hotel in York, Pa. Further details can be obtained by writing M. E. Dougherty, Eastern Regional Vice-President (see ASLE directory printed elsewhere).

### Chicago Practical Lubrication Course Enrolls 110

The Chicago Section has completed the practical course in lubrication which ran for 6 weeks during the months of October and November. Of the 110 men enrolled for the course approximately 40 have become new members of ASLE.

### Plant Maintenance & Engineering Conference

A maintenance and engineering conference will be held in conjunction with the 8th Plant Maintenance & Engineering Show to be held at Cleveland's Public Auditorium January 28, 29, 30 and 31. Included in the extensive program of maintenance practices are: Monday, January 28 — **Changing Requirements for Maintenance** (10:30 AM). Monday, January 28, 7:00 PM — (Concurrent Sessions) **Preventative Maintenance — A Case Report; Maintenance Cost Controls; Maintaining Handling Equipment; Maintenance of Equipment Standards;** and the **Role of Lubrication in Preventative Maintenance.** Scheduled for Tuesday, January 29 at 9:30 AM are the following concurrent sessions: **Organization — A Case Report; Yardsticks by Which to Measure Effectiveness of Maintenance; Training and Education for Maintenance Personnel; Maintenance in Metal working Plants; Maintenance in Chemical Plants; Maintenance in Steel Mills; Maintenance in Metal Fabricating Plants;** and **Mainte-**

**The recent Practical Lubrication Course held by the Chicago Section attracted 110 students from Midwestern industries.**

nance in Foundries. At 3:00 PM on Tuesday the following sessions are to be held concurrently; **Maintenance in Continuous Operation Plants; Maintenance in Paper Mills and Product Plants; Maintenance in Petroleum Refineries; Maintenance in Rubber Mills and Product Plants; Maintenance in Food Processing Plants;** and **Maintenance in Textile Mills.** On Wednesday at 9:30 AM all Monday evening sessions will be repeated concurrently, and on Wednesday at 7:00 PM the four sessions listed previously for Tuesday evening will be repeated concurrently.

Exhibits representing 400 companies will be shown during the four days of the concurrent maintenance and engineering show.

### Industrial Engine Oil Manual Announced

The Internal Combustion Engine Institute has recently released the handbook "Lubricating Oils for Industrial Engines" listing the brand names of oils meeting the requirements for three types of increasingly severe service.

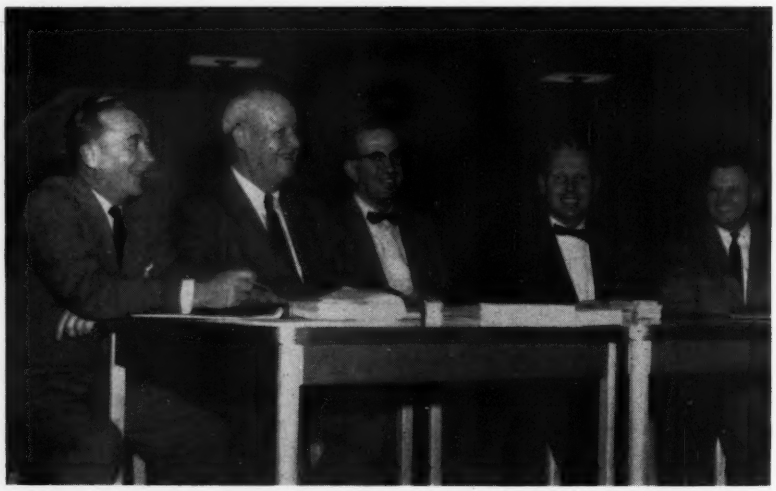
No evaluation of the relative merits of a particular product with a competitive product is attempted. The oils are grouped according to their ability to meet service requirements common to industrial engine operation. Lubricants for less severe service, such as motor car service, light industrial operations, etc. are outside the scope of the manual.

Trade names of the listed products and their performance levels are based upon information received directly from their respective suppliers. The book's 56 pages of listings include most of the foreign and American brand names of internal combustion engine lubricating oils.

Copies of the manual can be obtained from the Internal Combustion Engine Institute, Room 914, 201 N. Wells Street, Chicago 6, Ill. Single copies are 25c each and orders for 500 or more copies are 20c each. Postage will be added to all orders received and express or truck transportation charges only are collect. International postal certificates used in payment are redeemable in the United States at 8c each.

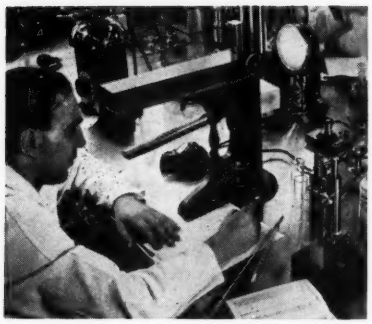
## ASLE JEWELRY

**The new ASLE jewelry is available in the form of tie clasps, lapel pins, or cuff links at $3.50, $1.50, and $4.50 each, respectively, from the National Office. Many of the Sections are using the jewelry in their membership campaigns. Details are available through the National Office.**



**Among the education committee members responsible for the Chicago Section-sponsored practical lubrication course were: J. N. Waddell, A. B. Wilder, C. W. Witham, C. A. Lorenzen, and P. I. Perkins.**

# Cover Story



**Laboratory Testing of Petroleum Products.** The photograph on the front cover of this issue, taken in a Cities Service laboratory at Cicero, Illinois, shows a skilled chemist engaged in making tests aimed at insuring that finished products fully conform to specifications. Such testing is the customer's guarantee that he will receive the quality product for the application intended.

A high percentage of the funds appropriated by petroleum companies for research and development is earmarked for projects aimed at keeping their products in pace with, or ahead of, changing industrial and automotive needs. This coordination of petroleum research with industrial planning has enabled the nation's manufacturers to advance at full speed while, at the same time, permitting them to maintain and, in some cases, to improve their basic supplier positions.

Regardless of all other aspects of any company's research program, fundamentally it must direct its interest toward the future welfare of the company itself, particularly with respect to ultimate profits. There are secondary values which include company prestige, advertising, contributions toward national security, and additions to the fund of human knowledge . . . but in the final analysis, a company's investment in research must justify itself in increased earnings or savings.

In the field of product development, the justification for work can be divided roughly into three categories; first, the improvement of existing products to keep pace with or exceed competition; second, the development of entirely new products to meet the requirements imposed by new or improved equipment and machinery; and third, the development of a product for which there is no known immediate use and for which a market has to be developed.

Research and development are similar to all operating activities where the degree of earning is a measure of the effort's success. Basically, however, the difference between the research and operating activities is one of time and application since operating expenditures normally return earnings almost immediately; but similar efforts, and dollars spent for research and development, require a longer time to reflect increased income.

To research and development management this time differential is vital, in that the research program as an entity must be thoroughly planned, executed, and supervised to prevent the passage of time from dulling the true evaluation of the ratio between expenditure and economic return.

At best, a dollars-and-cents evaluation of any company's research activities can be no more than an informed estimate, for many of its gains are of a nature which cannot be easily ledgered. The fact, however, that our major companies are now in the process of expanding their research activities indicates a growing recognition of the fact that sound research programs are not simply philanthropic aids to the progress of the nation, but a profitable investment for the companies which sponsor them. It is this, alone, which explains why today the American petroleum industry employs more than 15,000 scientists and technicians in their laboratories, and backs them up with appropriations which exceed $100,000,000 a year. (Submitted by Cities Service Co., Inc.)

# Coming Events

**JANUARY 28-31,** Plant Maintenance & Engineering Show, Public Auditorium, Cleveland, Ohio.

**FEBRUARY 18-22,** American Society of Civil Engineers (Convention), Jackson, Miss.

**MARCH 18 & 19,** Steel Founders' Society of America (55th Annual Meeting), Drake Hotel, Chicago, Ill.

**25-27,** American Society of Tool Engineers (25th Technical Meeting & Convention), Shamrock Hilton Hotel, Houston, Texas.

**APRIL 8-12,** American Welding Society (National Spring Meeting), Hotel Sheraton, Philadelphia, Pa.; April 9-11, Fifth Welding Show, Convention Hall, Phila.

**9-11,** The Canadian Institute of Mining & Metallurgy (Annual General Meeting), Quebec City, Quebec, Canada.

**15-17,** American Society of Lubrication Engineers (12th Annual Meeting & Exhibit), Sheraton-Cadillac Hotel, Detroit, Mich.

**16-18,** National Petroleum Association (Semi-annual Meeting), Hotel Cleveland, Cleveland, Ohio.

**29 thru May 3,** American Material Handling Society (7th national Materials Handling Exposition), Convention Hall, Philadelphia, Pa.

**MAY 5-11,** Second Cuban Petroleum Congress (Industry Exhibit, and Technical & Scientific Sessions), Havana, Cuba.

**JUNE 3-7,** American Society of Civil Engineers (Convention), Buffalo, N. Y.

**11-13,** Western Plant Maintenance & Engineering Show, Civic Auditorium, San Francisco, Calif.

**SEPTEMBER 11-13,** National Petroleum Association (Annual Meeting), Traymore Hotel, Atlantic City, N. J.

**23 & 24,** Steel Founders' Society of America (55th Fall Meeting), The Homestead Hotel, Hot Springs, Va.

**OCTOBER 7-9,** ASLE-ASME 4th Annual Lubrication Conference, Royal York Hotel, Toronto, Ont., Canada.

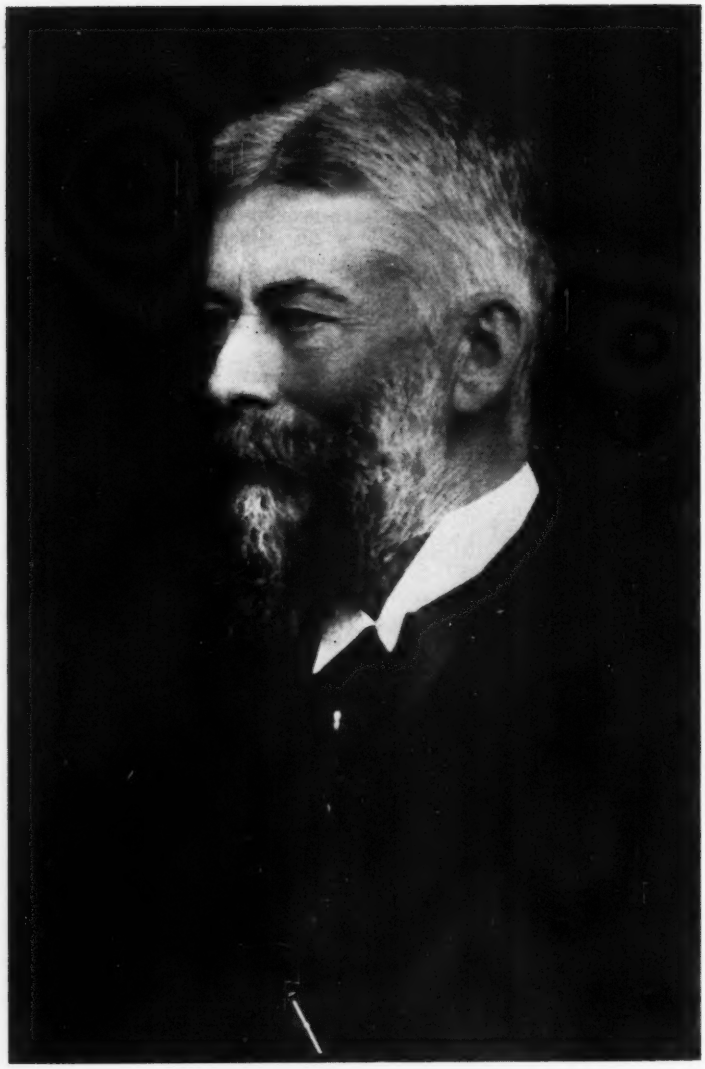
**14-18,** American Society of Civil Engineers (Convention), New York City.

**NOVEMBER 11-13,** Steel Founders' Society of America (12th Technical & Operating Conference), Carter Hotel, Cleveland, Ohio.

**13-15,** American Standards Association (8th National Conference on Standards, and 39th Annual Meeting), San Francisco, Calif.

## *Of Special Interest:*

**Professor George Hugel,** Head of the French Institute of Petroleum, Paris, will present his paper entitled "Chemical Nature of Extreme-Pressure Lubricants" at the **12th ASLE Annual Meeting & Exhibit.**



Beauchamp Tower, 1845-1904

# Men of Lubrication

by F. R. Archibald

**BEAUCHAMP TOWER.** There are many well known instances where scientific results of the most profound significance were prompted by the observation of apparently extraneous phenomena in the course of an experiment. The discovery of the phenomenon of film lubrication was due to such an irrelevant occurrence in some experiments on journal bearings. During the years 1882-83, Beauchamp Tower, an English engineer, was engaged on a series of friction experiments on journal bearings for the Institution of Mechanical Engineers. In order to test the effect of lubricating with an oiler, a hole was drilled in the bearing brass. When the oiler was not in place, it was found that oil streamed out through the hole and, to prevent this nuisance, the hole was stopped with a wooden plug. After a short time of running, the plug was forced out of the hole which was a clear indication of the existence of fluid pressure. Measurement with a pressure gage confirmed the fact of a pressure in the oil film.

Beauchamp Tower was born in 1845 at Morton Rectory, Essex, where his father was rector. His early engineering training was obtained at the Armstrong Works, Elswick, and later at the Tyne Iron Works where he was engaged in the design of steel ships. In 1869 he became assistant to William Froude, famous for his pioneer work on ship resistance. There seems to have been a great friendship between these two men, for in 1877 when Froude's health broke down from overwork, he asked Tower to accompany him on a voyage to the Cape of Good Hope. Froude did not survive the trip, and Tower returned to England in 1878.

In 1875 Tower met Lord Rayleigh and the two became close friends. At this time Tower had established a reputation as a careful experimenter, and Rayleigh asked him to become his laboratory assistant which he did for a few months. It is quite likely that Tower's connection with Rayleigh had a bearing on his being selected to perform the friction experiments for the Institution of Mechanical Engineers, because Rayleigh was a member of their committee on friction for whom the work was done. Lord Rayleigh's interest in the subject of hydrodynamic lubrication was, no doubt, due in some measure to his friendship with Tower. Tower's work with Rayleigh was mainly on some hydrodynamic experiments.

During the period of his work with Froude and, also, during his period with Lord Rayleigh, Tower worked privately on a scheme to drive ships by wave power. The idea was to install such a device in sailing ships to enable them to get through the doldrums where there is always a long swell. Tower's paper, "On a Method of Obtaining Motive Power from Wave Motion," was presented before the Institution of Naval Architects during 1875. The paper makes some rather impressive claims for the power possibilities which, if realized, could have been of value to the sailing ships of the day. However, the scheme does not seem to have worked out, possibly because of the expense and great bulk of the apparatus.

On his return from the Cape of Good Hope in 1878, Tower started to practice engineering on his own account, and it was during this period he developed his so-called spherical engine. At that time, one of the great problems of mechanical engineering was to devise a steam engine suitable for direct coupling to dynamos which required a high speed of rotation. English engineers were convinced of the soundness of direct coupling, and many engines having the desired high speed were designed. It appears that Tower's engine was the best of the contemporary designs. A company was formed to build and market the engines and a fair number of them were put in service.

Tower's work on the Friction Experiments for the Institution of Mechanical Engineers and his report of 1883 disclosed the existence of fluid films between a journal and its bearing when the oil supply was copious. The accidental discovery of the phenomenon was immediately followed by a careful investigation of the pressure at various points in the fluid film. These pressures, when integrated over the area of the bearing, gave a result in close agreement with the load being carried. The hydrodynamic nature of lubrication was then clearly demonstrated. Tower was apparently surprised to find that the resistance to motion was not proportional to the square of the sliding velocity but, rather, to its first power. The work of Darcy in France, on the flow of water through pipes, had shown a resistance to flow proportional to the square of the velocity. The matter was cleared up by Osborne Reynolds, who showed experimentally that a fluid in motion can give resistance proportional to the first power of velocity or to the square of the velocity, depending on whether the flow is laminar or turbulent. Reynold's great paper of 1886, "On The Theory of Lubrication," put Tower's findings in mathematical form and enabled the important subsequent developments in bearing design to be made.

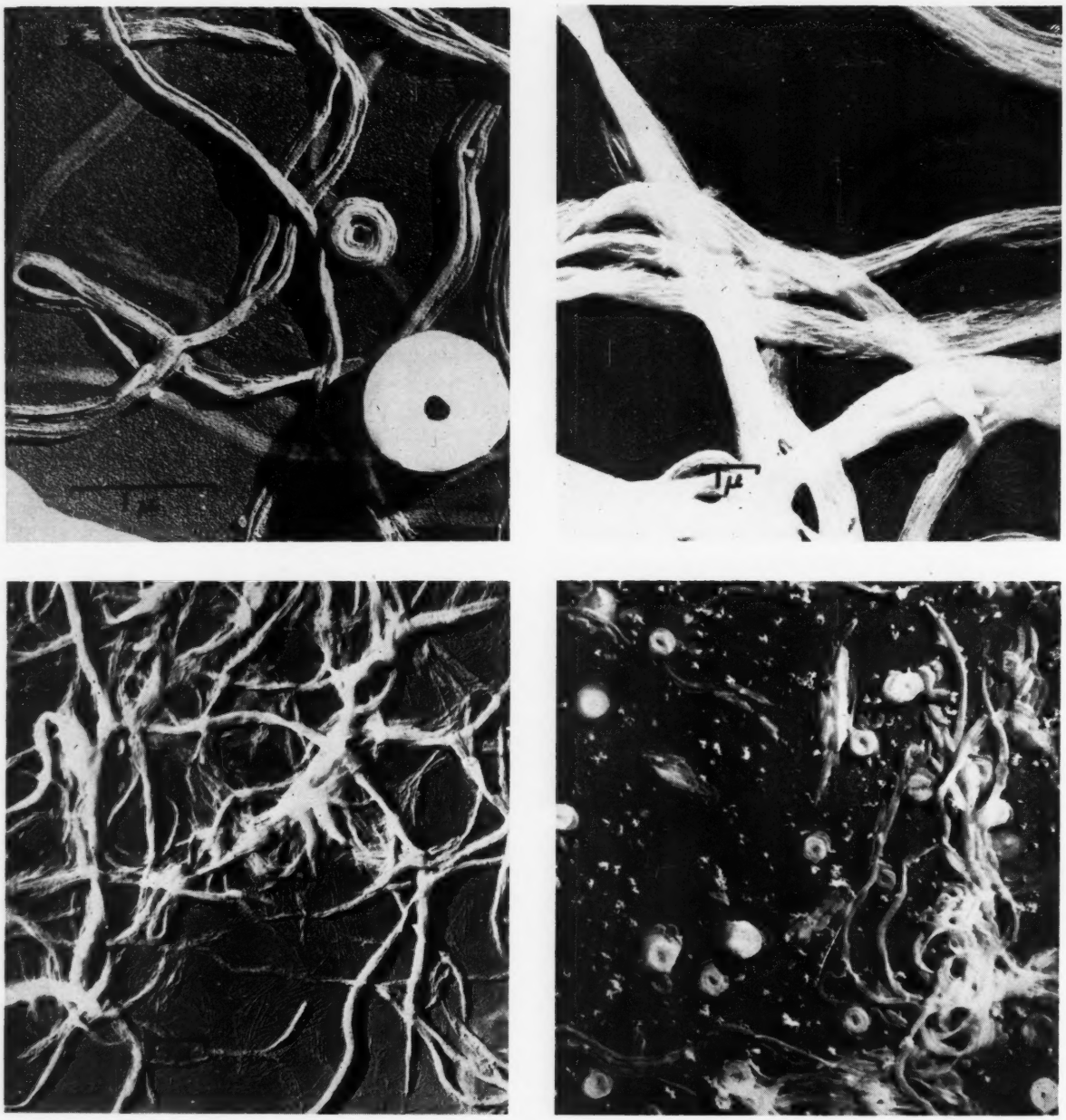
The later years of Tower's life were embittered over the failure of the Admiralty to adopt his "steady platform" for guns and searchlights at sea. By means of gimbols mounting and using a gyroscope, Tower devised a steady platform which would maintain its position within half a degree. The Admiralty had encouraged Tower to proceed with this extensive work of development but, in spite of the success of the device when tested, they decided that it was not suitable for the service. The main reason appears to have been that the additional weight of the apparatus could more usefully be employed in

(Continued on p. 63)

# Lubrication in the News

## SOAP FIBERS IN LUBRICATING GREASES

In the wonderful world of electron microscopy, matter is revealed in such fantastic forms as shown by these micrographs of soap fibers in lubricating greases (magnification: 25,000 diameters). In studies of lubricants, fuels, catalysts, and metal surfaces, particles too small to be seen with the light microscope are examined, and significant scientific conclusions reached by relating physical properties and the images seen. Electron microscopy is based on theories concerning the behavior of high velocity electrons, and the discovery that a magnetic field can be used as a lens to focus a beam of electrons. High resolving power in such a microscope clearly reveals the shape and structure of particles. Preparing specimens and creating three-dimensional impressions test the imagination and ingenuity of the microscopist. (Submitted by The Texaco Research Center, Beacon, N. Y.)



Electron Micrographs of soap fibers in new (top) and used (bottom) greases, magnified 10,000 to 25,000 times.

'dag' dispersions . . . *a touch does so much!*

# 'Oildag' vital to continuous molding-machine operation

Production Pattern & Foundry Company of Chicopee, Mass. runs its five shell-molding machines 24 hours a day to keep up with casting-floor demand for molds. To stand up under this tough service, the machines must be properly maintained.

Maintenance personnel at Production Pattern & Foundry have found that lubrication instructions of the machine's manufacturer have paid off handsomely. Shell Process Co., which makes these shell-molding machines, recommends **'Oildag'** — colloidal graphite in petroleum oil — diluted with five parts of No. 10 oil for the lubrication of critical wear points: the axle bearings of the drive mechanism, and hitch-pivots and link pins of the roller chain. Since the roller chain is subjected to 900° F temperatures within the shell-baking oven, conventional lubricants would be useless . . . would decompose and carbonize. Proof of the effectiveness of colloidal graphite for this heavy-duty high-temperature lubrication job is the fact that the machines have operated without breakdown for more than six years!

Dispersions of colloidal graphite are extremely valuable elsewhere in the foundry too: as long-lasting lubricants for moving parts of casting machines and other foundry mechanisms; as heat-resistant ladle coatings, and effective mold coatings for better parting and controlled cooling. You can learn more about the use of **'dag'** dispersions in metal-casting operations through Acheson's Bulletin 425. Write for your free copy today.

**ACHESON COLLOIDS COMPANY**
**PORT HURON, MICHIGAN**
. . . also Acheson Colloids Ltd., London, England

ACHESON COLLOIDAL DISPERSIONS:
Graphite • Molybdenum Disulfide • Zinc Oxide • Mica and other solids

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**ACHESON COLLOIDS COMPANY**
**Port Huron, Michigan, Dept. G-1**

Yes, I want your free bulletin describing 'dag' Dispersions for High-Temperature Lubrication.

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# Section News

**CLEVELAND, November meeting.** K. D. Reed (ASLE Pittsburgh Section), Dravo Corp., presented a paper entitled "Continuous-Flow Oil Circulating Systems" covering the various systems in use and the application of various types of pumps, valves, and filters. R. F. McKibben (ASLE Central Regional Vice-President), National Cash Register Co., and A. H. Turner (ASLE Cleveland/Youngstown Area Representative), Standard Oil Co., were guests of honor. (Submitted by G. L. Smith, Sec'y.)

**DULUTH/IRON RANGE, October.** Dr. E. R. Booser (ASLE President), General Electric Co., presented a paper entitled "Atomic Power Plants Lubrication."

**December.** "Automatic Lubrication Systems," by J. Jenkins, Industrial Sales Manager, Lincoln Engineering Co. (Submitted by C. D. Johnson, Sec'y.)

**FORT WAYNE, November.** Annual Social Night, featuring the showing of the 1956 Indianapolis 500-Mile Race film, followed by a discussion by P. M. Dabney, Socony Mobil Oil Co., and concluded with a tour of the Falstaff Brewery. (Submitted by C. R. Griffith, Chrmn.)

**HOUSTON, December.** Election of Officers (see ASLE Directory). (Submitted by W. S. Nicholson, Sec'y.)

**KANSAS CITY, November.** J. R. McCoy (ASLE Indianapolis Section), Indiana Commercial Filters Corp., presented a paper entitled "The Filtration of Industrial & Lubricating Oils."

**December.** Annual Christmas Party, featuring color movies of Portugal, Maderia, and rural England taken and presented by A. E. West, A. E. West Petroleum Co. (Submitted by J. D. Jenkins, Sec'y.)

**MONTREAL, October.** P. Kingsmill, Sales Manager of Peckover's Ltd., presented a paper entitled "Fitting Nylon to Industry."

**November.** "Fire-Resistant Hydraulic Fluids," by H. Zbornk, Celanese Corp. of America.

**December.** "The Filtration of Industrial & Lubricating Oils," by J. R. McCoy, Indiana Commercial Filters Corp.

**1957 Program Schedule:** Jan. 2, Open Forum; Feb. 6, "Anti-Friction Bearing Maintenance," by P. B. French, Lyman Tube & Bearings, Ltd.; Mar. 6, "The Frictional Resistance of Oils & Greases," by A. G. Rogers, Imperial Oil Ltd.; Apr. 3, "The Lubrication Engineer in Modern Industry," by C. L. Pope, Eastman Kodak Co.; May 1, "Lubricant Development," by A. L. Moore, Shell Oil Co. of Canada, Ltd. (Submitted by K. Bloxham, Sec'y.)

**PITTSBURGH, October.** E. E. Smith, Chemical Development Div., Climax Molybdenum Co., presented a paper entitled "Molybdenum Disulfide in the Field of Lubrication."

**November.** "Preventive Maintenance and Lubrication," by C. J. Wyrough, Pittsburgh Works Div., Jones & Laughlin Steel Corp. (Submitted by A. J. DeArdo, Publicity Chrmn.)

**TWIN CITIES, December.** Past Chairmen's Night, featuring J. Jenkins, Lincoln Engineering Co., as guest speaker. (Submitted by C. H. Sweet, Sec'y.)

**WHEELING, September.** W. A. Howe, Gulf Oil Corp., presented a paper entitled "Diesel Oils & Fuels."

**October.** "Air Compressor Design & Maintenance," by B. Dunham, Sun Oil Co. (Submitted by S. J. Litten, Sec'y-Treas.)

# Foreign Lubrication Abstracts

(This is the ninth in a series of abstracts of translations of Foreign articles on lubrication, by Henry Brutcher. Copies of the full translations are available, at the indicated prices, by writing Mr. Brutcher at P. O. Box 157, Altadena, Calif.)

**On the Types of the Wear Process under Conditions of Dry Friction,** by I. V. Kragel'skii & E. M. Shvetsova; "Doklady Akademii Nauk SSSR," Vol. 75, No. 5, 1950, pp. 681-683; 3 figs., 1250 words. Fundamentals of wear. Definition of variables in sliding wear. Formulas. Two external factors governing sliding wear. Three basic types of interaction between surfaces sliding against each other. Study of pairs of like metals for wear. Data on wear of 0.80% steel at different speeds of sliding and under different pressures. Critical deformation as function of sliding speed for thermal effects and tearing-out effects. Most important conclusions drawn from authors' result for minimizing wear in actual service. Unfortunately, no information on testing procedure is given by the authors, and their results are in part at variance with experience in this country. (Order No. 2678, price $2.50)

**On the Problem of the Wear Resistance of Chromized Surfaces,** by N. S. Gorbunov & V. P. Lazarev; "Doklady Academii Nauk SSSR," Vol. 86, No. 2, 1952, pp. 345-347; 3 figs., 1 table, 850 words. Experimental study of wear resistance of chromized surfaces on iron. Results of microhardness tests on chromized Armco iron. Particulars on wear testing arrangement developed by Deryagin and Lazarev (Fig. 2). Wear test results (Table I & Fig. 3) obtained on unchromized vs. chromized iron. (Order No. 2966, price $1.70)

**Abrasive Wear of Metals at Various Temperatures & Speeds,** by G. I. Kiselev; "Doklady Akademii Nauk SSSR," Vol. 87, No. 5, 1952, pp. 735-737; 4 figs., 1500 words. Investigation of relationship between abrasive wear of metals and their tensile strength, at various temperatures. Experimental procedure. Materials studied: plain carbon steels (0.20, 0.45, 0.60, and 1.20% C); copper, and zinc. Difference be-

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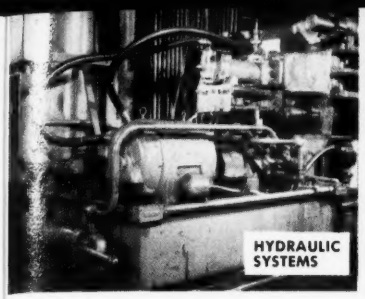


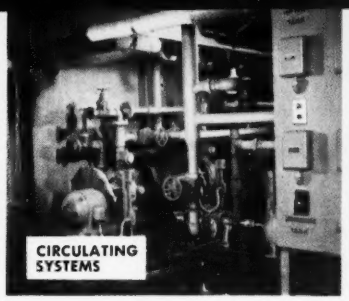

# SUN SOLNUS OILS IDEAL LUBRICANTS FOR 80% OF ALL APPLICATIONS

**Moderately priced...low in carbon-forming tendencies,** Sun Solnus® oils simplify your storage problems by doing with *one oil* many jobs that would otherwise require *several.* Their ability to protect metal parts against corrosion, their resistance to oxidation, and their moderate price all add up to "more lubrication per dollar."

For technical information, see your Sun representative, or write to SUN OIL COMPANY, Philadelphia 3, Pa., Dept. I-51.

SUNOCO®

INDUSTRIAL PRODUCTS DEPARTMENT

**SUN OIL COMPANY**

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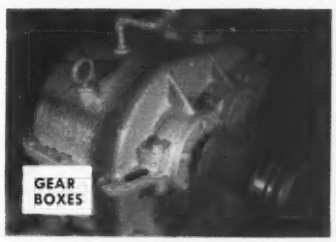


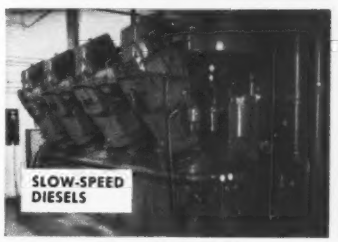

Valve on left is from compressor run for 3,000 hours with well-known, high-grade oil. At right is same valve after a 3,000-hour run with Sun Solnus oil. Note difference in carbon deposits.

# 3000-HOUR TEST PROVES SUN SOLNUS OILS REDUCE CARBON BUILD-UP IN COMPRESSORS

**Equipment:** A three-stage Norwalk horizontal-type compressor. Operating pressure: from 1,000 to 1,500 psi.

**Test:** The compressor was cleaned thoroughly and filled with a well-known, high-grade oil. The equipment was run for 3,000 hours, then torn down for inspection and cleaning. Then Solnus® 300 was tested in the same way.

**Results:** Look at the two pictures. You can see for yourself how Solnus oil reduced dangerous carbon build-up.

All types of reciprocating air compressors that have been changed over to a Sun Solnus oil show similar results. A test in your compressor will show the same remarkable reduction of carbon deposits.

You can get a technical bulletin about Sun Solnus oils by asking your Sun representative, or write to SUN OIL COMPANY, Philadelphia 3, Pa., Dept. I-52.

SUNOCO®

**INDUSTRIAL PRODUCTS DEPARTMENT**

**SUN OIL COMPANY** PHILADELPHIA 3, PA.

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# Lube Lines

By A. F. Brewer*

## SIGNIFICANCE OF TESTS

The tests which may be carried out on any lubricant to determine lubricating value and adaptability to service conditions, may be physical or chemical laboratory tests, or service tests run on actual operating machinery either in the field or laboratory. Any type of test has a certain amount of significance; sometimes it relates directly to service performance of the lubricant, sometimes it is irrelevant. The American Society for Testing Materials in the 1955 edition of *Significance of ASTM Tests for Petroleum Products* lays considerable emphasis upon this question of significance and the relative value in predicting the probable results (lubricationwise) which may be expected.

Management is not too interested in the laboratory procedure details of any of the conventional tests, but they are vitally concerned with the significance of these tests. By being fully informed in this regard, they can make sure that their purchasing people are considering only those tests which relate directly to expected service performance of the lubricants which they are buying. By concentrating only on these tests and disregarding consideration of tests such as gravity and color which so often are irrelevant, they can simplify their purchasing procedure and be most certain that their lubricant supplier is quoting on products best suited both physically and chemically for the intended service.

Lubricating value is of primary importance. The tests which may indicate this in a turbine oil would not necessarily apply to a heavy-duty Diesel engine oil. By the same premise, the tests which would indicate the stability and resistance to oxidation of a grease for precision ball or roller bearings might be secondary for a steel mill grease where load-carrying (extreme pressure, EP) characteristics must be high.

Stability as related to lubricating value is one of the most important properties in practically any lubricant; in a product intended for a circulating or hydraulic system, or for pressure lubrication in a totally enclosed system, stability as indicated by resistance to oxidation or chemical decomposition must be as high as possible. Here the ability of the additive or inhibitor to impart resistance to oxidation should be of primary concern to management, assuming that the consistency of the grease or viscosity of the oil under consideration is suited to the prospective operating temperatures.

In studying the significance of any laboratory test for oxidation, the service conditions under which the lubricant may have to function must be considered. A test for oxidation-resistance of a turbine or hydraulic oil differs in procedure from a test for internal combustion engine oil. The same holds true for oxidation tests for greases. As yet no universal test for oxidation of lubricants has been perfected. By correlating results from any one of the special tests now used, with the performance ability of the lubricant to which the test is adapted, a valuable picture of what can be expected in long-time service can be developed.

In a turbine oil the rate of change of the neutralization number is significant with respect to progress of oxidation. Provided the oil is relatively free from outside contamination, any sudden rise in the neutralization number should be taken as a warning that some condition prevails which is inducing oxidation and impairing the lubricating value of the oil. The interfacial tension test, also, has recently been given quite some attention as an indication of when a lubricant such as turbine, hydraulic, or circulating oil has been so reduced in lubricating value by reason of the presence of water-absorbing compounds as to render it no longer serviceable. In brief, oxidation and the presence of oxidation products reduces the interfacial tension of such an oil.

In studying a grease for durability and resistance to oxidation, it should be subjected to the expected operating conditions. This can be done in a test machine in which the bearing and its charge of lubricant is at least partly visible, thereby enabling constant observation of the behavior or action of the lubricant. It is particularly important to note the extent to which thickening or separation develops, because these conditions along with oxidation are a significant indication of breakdown and reduction in lubricating value. This type of test has added significance in that under carefully controlled laboratory conditions of temperature and speed, the limits as to these latter can be established.

Coupled with information as to stability, data as to pumpability also can be highly significant. Here the penetration test is important, since it is a valuable guide as to the probable flow characteristics of a grease, how it will function in a centralized lubrication system, and the ease with which it can be handled especially in cold weather. In effect, as the ASTM states: "A grease must function as a viscous fluid as soon as it is placed in service."

*Consultant, and Author of *Basic Lubrication Practice.*


*Memo*

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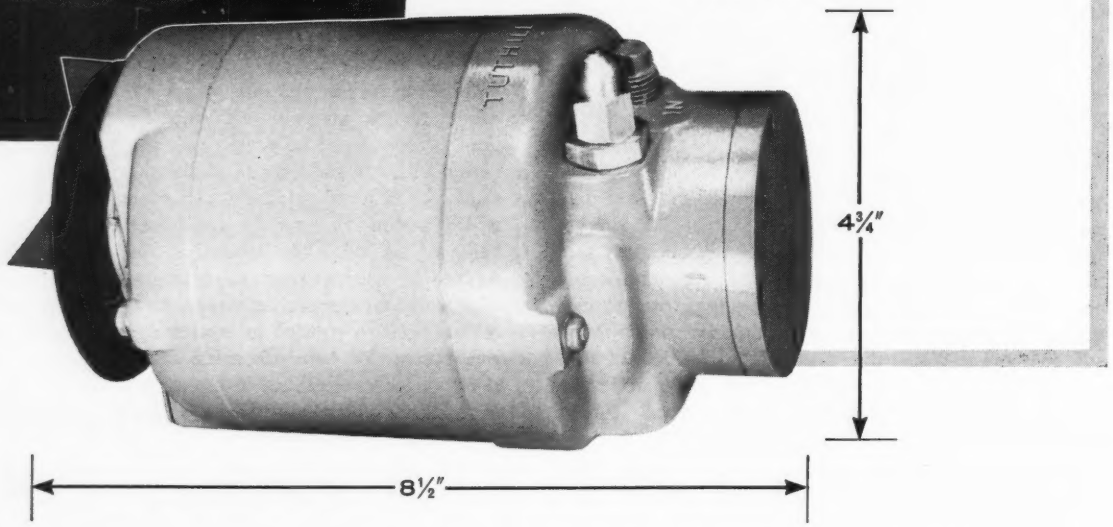


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PUMPS FOR YOUR PURPOSE

# The Scope of the Problem of Occupational Dermatoses Due to Cutting Oils

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An occupational dematosis is defined as any abnormality of the skin resulting from, or aggravated by, the industrial environment. This definition covers the entire group of occupational skin diseases, one type of which is caused by cutting oils.

According to available statistics, about two-thirds of all reported compensable occupational diseases fall into the category of occupational dermatitis. Actually, this figure does not represent the total incidence of occupational skin disease, because many workers who develop dermatitis do not necessarily lose time and, therefore, are not classified as compensation cases.

According to the 1950 Census of Population, there are approximately 529,000 machinists employed in the United States. It can be assumed that a large portion of these come into contact with a variety of cutting oils used in their trade. Information based on statistical reports and clinical examinations shows that cutting oils are the most frequent causes of dermatitis among machinists and allied tradesmen. This is not surprising, for the use of cutting oils is widespread in all kinds of industries, especially manufacturing plants where they are handled by one or more departments.

In 1943, a five-year statistical compilation of causes of occupational dermatitis showed that 18.8% of 41,000 cases were attributed to contact with petroleum products and greases. In 1952, incidence statistics gathered from eleven states showed that 19% of 6,700 occupational dermatitis cases were due to petroleum products. In both studies, solvents and thinners were classified apart from petroleum products, but undoubtedly they included cases due to contact with cutting oils.

D. J. Birmingham

These incidence figures may not coincide with those in plants where good safety, medical, and industrial hygiene practices are common. On the other hand, when one realizes that fully 70% of the working population does not have access to good in-plant

This paper was presented to the ASLE 10th Annual Meeting, Chicago, April 13, 1955.

## EDITOR'S NOTE

The inclusion of a Symposium on Industrial Dermatoses at the ASLE 10th Annual Meeting held in Chicago on April 13, 1955, represented a rather radical departure from the normal fields covered by the Society. However, it was felt that this subject was of such vital concern to so many people that it deserved a place on the meeting program. The tremendous interest displayed and the great success of the symposium was fitting testimony to the wisdom of making it a part of the program.

The panel was organized by Dr. C. S. Livingood (Physician-in-Charge, Division of Dermatology, Henry Ford Hospital, Detroit), and was composed of specialists in their respective fields including: Dr. D. J. Birmingham (Chief Dermatologist, Occupational Health Program, U. S. Public Health Service, Dept. of Health, Education & Welfare, Cincinnati), M. L. Beardslee (Resident Engineer, Wayne Plant, Detroit Diesel Engine Division, General Motors Corp.), Dr. E. A. Irwin (Medical Director, Ford Motor Co.), N. H. Schell (Lubrication Engineer, Fort Wayne Works, International Harvester Co.), Dr. J. M. Shaw (Instructor in Dermatology, University of Michigan, School of Medicine), Dr. L. F. Weber (Clinical Professor of Dermatology, University of Illinois), and Dr. W. C. Witham (Program Coordinator, Department of Chemistry & Chemical Engineering, The Armour Research Foundation). Each panel member was selected because of his special interest, experience, and expert knowledge of the various phases of the problem.

Through the efforts of Dr. Livingood, the contributions of the panel have been converted into a form suitable for publication in *Lubrication Engineering*, and will be presented as a series in consecutive issues of the Journal. The first paper, entitled "The Scope of the Problem of Occupational Dermatoses Due to Cutting Oils," by Dr. Birmingham, appears above. Other papers in the series which will appear in succeeding issues are: "Chemical Composition of Lubrication Oils, Cutting Oils, or Petroleum Solvents" (Dr. Witham), "Skin Diseases Caused by Cutting Oils" (Dr. Weber), "The Prevention of Cutting Oil Dermatoses" (Dr. Birmingham), "Prevention of Industrial Dermatitis in a Motor Truck Plant" (Mr. Schell), "Prevention of Industrial Dermatitis" (Mr. Beardslee), "The Composition & Action of Protective Ointments" (Dr. Shaw), and a Transcript of the Question & Answer Period.

medical services, it can be readily understood why the occurrence of dermatoses is not lessening, and why after ten years the proportion of cases due to petroleum products is practically the same. The dermatoses hazards in the small and the large machine shops are the same. They differ only in that they generally persist unchecked and even unrecognized in the absence of good safety, medical, and industrial hygiene practices.

For better understanding of the scope of this problem, one must appreciate the numerous factors which influence the inception of an occupational skin disease. Besides direct causes, which in this instance happen to be cutting oils, there are a variety of indirect or predisposing factors which should be mentioned. These are as follows:

1. RACE.

Certain races differ in the color and texture of their skin. These racial variations may provide an individual with a more protective, or at times, a more vulnerable type of skin. For example, the white man is less able to withstand the effects of sunlight and coal tar products than the Negro. Conversely, the Negro is definitely more prone to form keloid or redundant scar tissue following minor injury.

2. TYPE OF SKIN.

The dark, swarthy, hairy, and oily skin, though better able to withstand the action of solvents, is predisposed to the development of occupational acne from oil. This is due to the already present highly active sebaceous or oil glands, and the numerous hair follicles which allow settling points for industrial contactants such as cutting oils. By the same token, the light complexioned, so-called "thin skinned" person is somewhat intolerant to the action of solvents, but is able to work with most of the cutting oils without great difficulty.

3. AGE.

Age, *per se*, is not an important factor; however, occupational dermatitis occurs frequently among young workmen because they are less inclined to adjust rapidly to hygienic practices within a plant. It often requires a distasteful experience with dermatitis to create a sufficient impression of the need of hygiene. Chronic skin disease is more common among old workmen, and as a result they are likely to change their jobs. This is perhaps due to the general lessening of skin "toughness" or resistance influenced by less skin oil, drier skin, poorer circulation of blood, and subsequent lessened recovery powers.

4. ALLERGY.

Only 20% of all occupational skin diseases is based upon allergy. Thus, almost 80% is due to the effect of primary irritants, and most of the cutting oils are classified as insidious or low-grade primary irritants.

5. LACK OF CLEANLINESS.

No predisposing factor is as important in bringing about an occupational dermatosis as the lack of cleanliness. It is a difficult problem because the human element is involved. While the clean worker is able to maintain good personal and plant hygiene during the course of his work, the careless employee may have no interest in maintaining cleanliness of his person or his shop. This factor must be vigorously met by education, adequate washing facilities, protective clothing, and a good control of the environment.

As direct causes, the cutting oils can be divided into two large groups — the insoluble* and the soluble oils. The insoluble compounds may contain a large variety of ingredients which help to make them satisfactory lubricants, aid the tools in the cutting operation, and keep the metals from corrosion. The soluble oils act as coolants and include the newer emulsions and synthetics so highly rated at this time. In our experience, the insoluble oils have been generally found to be more offensive to the skin of most workmen than the soluble compounds. They tend to create several types of lesions associated with the action of petroleum oils and additives upon the human skin. The soluble oils do not attack the skin in the same manner. They differ in structure and behave somewhat like a soap in solution. Although they can produce dermatitis, they are generally less offensive than the insoluble oils.

Research and development on cutting oils has led to many advances in the lubricant field. Some manufacturers, however, during the developmental phase in the laboratory, do not consider the potential harmful effects of cutting oils. This omission has added to the scope of the cutting oil dermatitis problem. It is reasonable to expect the responsible manufacturer, when producing better cutting oils, to also study the effects of these materials upon a working population which must handle them.

Finally, while it is recognized that occupational dermatoses occur in large numbers, it is of equal importance to understand that not all skin afflictions which attack the working man are occupational in origin. Unless a reasonable degree of fairness is shown in this regard by both employer and employee, the problem becomes further complicated. Like the practice of good hygiene, the recognition of these cases must be based upon a knowledge of the materials being handled and what they can do to the skin.

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*This refers to mineral oils, either with or without additives, as differentiated from the so-called soluble oils, which are of mineral oil base but are mixed with water to form an emulsion before use.


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# Grinding Temperatures

**The several types of temperatures that are of interest in grinding operations are discussed, and it is concluded that the temperature at the very surface is the one of major interest with regard to crack formation and surface damage. Apparatus is described for measuring the temperature of a freshly ground surface by means of a photoconducting lead sulfide cell and an oscilloscope. The experimentally observed surface temperatures were found to vary in the same way with chip size as calculated temperatures based on specific grinding energy. Both water- and oil-base fluids were found to reduce surface temperatures significantly over dry grinding. Oil-base materials were found to give slightly lower temperatures than water-base materials, due to the superior lubrication action of the oils more than offsetting their relatively poorer cooling action.**

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## INTRODUCTION

In metal cutting operations one is interested in tool tip temperatures, since the life of the cutting tools is influenced in an important way by these temperatures. Studies of many cutting operations have shown that tool tip temperature varies as the square root of the cutting speed. The grinding operation is characterized by high cutting speeds (usually about 6000 fpm) and, hence, surface temperatures that are high relative to lathe tool temperatures are to be expected. The tool materials that are used in grinding operations (aluminum oxide and silicon carbide) are far more refractory than those ordinarily used in machining and, hence, are capable of operating at higher temperatures without softening. Furthermore, since in a grinding operation a given abrasive particle is at maximum temperature for an extremely short time, temperatures much higher than those that could be sustained successfully under equilibrium conditions may be employed. Normally, temperatures in grinding are more of interest with regard to surface damage than wheel wear, although in certain special cases, such as the grinding of titanium alloys or vanadium containing tool steels, wheel wear becomes the controlling problem. Tarasov[1-4] has considered the influence of grinding variables on workpiece damage in a series of papers.

In any nonsteady state, a combination of time and temperature, rather than process temperature alone, is of interest. This fact makes it possible for metals to reach temperatures far in excess of their equilibrium melting points without melting, provided the time at temperature is short enough. It is immediately evident that the meaning of high fleeting temperatures with regard to physical and chemical changes of a metal surface is difficult to interpret. The only manner in which grinding temperatures can be used, at the present time, is in a relative way. While we cannot say that a certain fleeting grinding temperature will lead to damage of the ground surface or abnormally high wheel wear, we can assure that these tendencies will be lessened when conditions are changed so as to reduce the peak grinding temperature. In this paper, a method of approximately measuring the temperature on a ground surface is presented and then applied to a study of the influence of different grinding fluids.

## TYPES OF GRINDING TEMPERATURES

In metal cutting, the standard method of estimating tool tip temperatures is the chip-tool thermocouple method[5-7] in which the dissimilar metals of the tool and chip constitute a hot junction in a thermoelectric circuit. While this method is not generally applicable in grinding due to the high impedance of a vitrified grinding wheel, it has been used with special silicon carbide wheels of relatively high conductivity.[8] In the early stages of the present study, satisfactory results were also obtained using a diamond wheel with iron bond in grinding sintered tungsten carbide. However, the latter system is rather far removed from the grinding of hardened steel with an aluminum oxide wheel, which is of greater commercial interest. Several attempts to clamp two dissimilar metals together in a horizontal plane and to measure the temperature peak thermoelectrically with an oscilloscope as the wheel passes over the junction, have not been successful. The idea behind these tests was that in the instant the wheel is over the junction, a chip would be generated that was in part one metal and in part the other.

McKee, Moore and Boston[9] studied the influence of grinding variables, including fluids, upon the general workpiece temperature after grinding as measured by a portable contact surface thermocouple. The values recorded in this work were quite low and have little relation to the values that are under consideration here. Littmann and Wulff[10] imbedded fine constantan wires in the workpiece and used the work itself as the other conductor. By following the

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This paper was presented at the ASLE 11th Annual Meeting, Pittsburgh, April 5, 1956.

increase in temperature on an oscilloscope, as successive layers were removed, it was possible to estimate maximum workpiece temperatures within about 0.001 inch of the surface.

Outwater and Shaw[8] considered the grinding operation from the point of view of a nonsteady state heat transfer problem, and arrived at the following expression for the mean temperature on the shear plane during grinding.

$$\bar{\theta}_s = \frac{u_s}{12J(C\gamma)} R \qquad (1)$$

where $R$ = the percentage of the total shear plane energy going into the chip; $u_s$ = shear energy per unit volume associated with the formation of the chip, in lb/cu.in.; $J$ = mechanical equivalent of heat, (778 ft. lbs./BTU); $C\gamma$ = volume specific heat of workpiece, BTU/in.$^3$/°F.

The total grinding energy per unit volume of metal removed ($u$, see Figure 1) has two components: that associated with the plastic deformation which occurs during chip formation ($u_s$) and that associated with friction between chip and work ($u_f$). It has been estimated that $u_s$ and $u_f$ are each approximately equal to $u/2$[11] and, hence, Equation (1) may be written:

$$\bar{\theta}_s = \frac{u}{24(778)(C\gamma)} R \qquad (2)$$

The quantity $R$ is given approximately as follows[8]

$$R = \frac{1}{1 + 1.33\left(\frac{K \cdot \cot \phi}{V't}\right)^{1/2}} \qquad (3)$$

where $K$ = thermal diffusity of the workpiece, in.$^2$/sec.; $\phi$ = the shear angle associated with grinding chip formation; $V'$ = the grinding wheel speed, in./sec.; $t$ = maximum undeformed chip thickness, in. (see Figure 1).

When values are substituted into Equation (1), very high grinding temperatures (> 3000°F. )are obtained.

Other basic studies of the grinding operation[11, 12] have revealed that the specific energy associated with grinding ($u$) is influenced mainly by the size of the chips generated ($t$), and Reference 13 discusses in detail methods of computing ($t$). Figure 2 shows diagramatically the manner in which ($u$) varies with the undeformed chip thickness ($t$), and three distinct regions are evident: Region I — large values of $t$, $u$ increases with decrease in $t$; Region II — $u$ independent of $t$; Region III — small values of $t$, $u$ again increases with decrease in $t$.

The increase in ($u$) with decrease in ($t$) for large

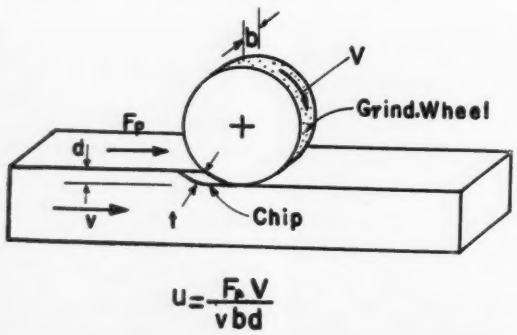



Figure 1. Surface grinding operation illustrating notation used.

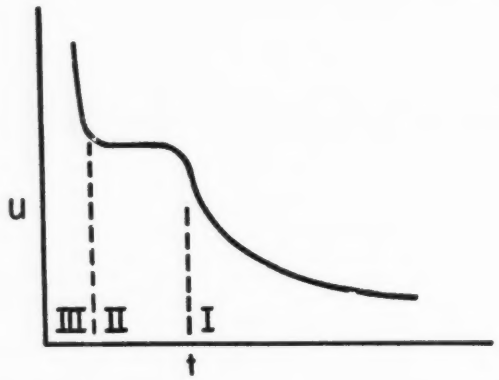



Figure 2. Variation of grinding energy per unit volume ($u$) with undeformed chip thickness ($t$).

chips and the subsequent leveling off of ($u$) has been associated with an increase of strength of the metal with decreasing size of specimen deformed (frequently referred to as the size effect), while the rise in $u$ observed in Region III is believed due to an increase in sliding friction for very small chips. From Equation (2) we should expect that when $V$ is constant, grinding temperature ($\bar{\theta}_s$) will vary in about the same way with ($t$) that $u$ does, since the quantities $R$ and ($C\gamma$) remain constant.

In the above discussion three types of temperatures have been mentioned: (1) that measured by McKee, Moore, and Boston with a contact thermocouple *after* grinding; (2) that measured by Littmann and Wulff *during* grinding with a thermocouple junction buried within 0.001 inch of the surface; (3) that calculated by Outwater and Shaw for the surface of the metal during grinding ($\bar{\theta}_s$).

It is evident that the values involved will increase in the order listed. The first temperature is not thought to be of importance with regard to wheel wear or surface damage, but such ambient temperatures are of importance in gaging. A part having a high ambient temperature after grinding will change in size upon cooling, and this constitutes an obstacle to the mass production of parts of high precision. The second temperature is thought to be of importance with regard to overtempering of a hardened workpiece and other subsurface damage initiating one or more 0.001 inch below the surface. The third temperature is the temperature at the surface during grinding, and is thought to be of importance with regard to grinding wheel wear and the formation of surface cracks in a hardened workpiece during grinding. It is the measurement of the third type of temperature we are concerned with in this paper and, more specifically, the influence that different types of grinding fluids have upon this temperature.

## RADIATION TEMPERATURE MEASURING TECHNIQUES

The surface temperature of a hot object can be determined by measuring the intensity of radiation from the surface. This is the principle of the optical pyrometer, and several years ago Schwerd[14] used a very sensitive bolometer with rock salt lenses to investigate the variation of temperature across the section of a metal cutting chip. More recently, the development of photoconducting cells of very short response time have made it possible to measure temperature flashes of very short duration by the radiation

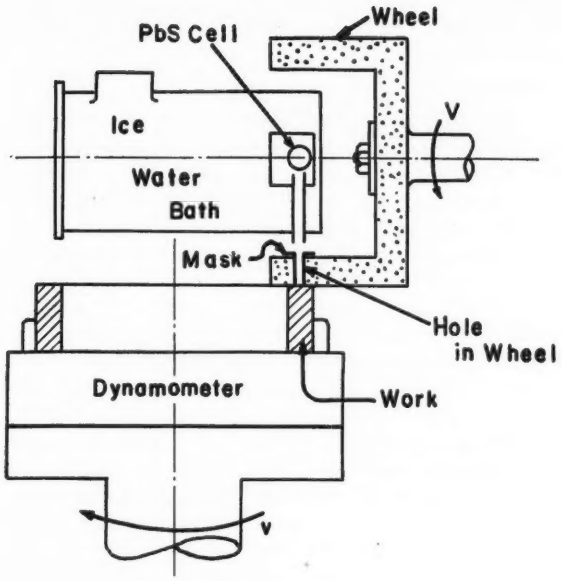



Figure 3. Apparatus.

method. These cells consist of a thin layer of lead sulfide deposited on the inner surface of a small glass envelope that is about the size of a pencil eraser. Two leads attached to opposite sides of the layer by printed circuit technique are joined to the prongs of the tube. When infrared radiation from the object under investigation falls upon the lead sulfide layer, its resistance changes, and this change in resistance can be measured conveniently.

Parker and Marshall[15] used a lead sulfide cell to investigate the surface temperatures obtained on car wheels during braking, and Marshall and Mackenzie[16] have applied the cell to produce a portable radiation pyrometer. Huggins, Roll, and Udin[17] have used such cells to measure transient surface temperatures during a spot welding, while Bowden and Thomas[18] found general agreement when comparing lead sulfide determined temperatures of friction sliders with those obtained by other methods. Details of lead sulfide cell characteristics may be found in References 19 and 20.

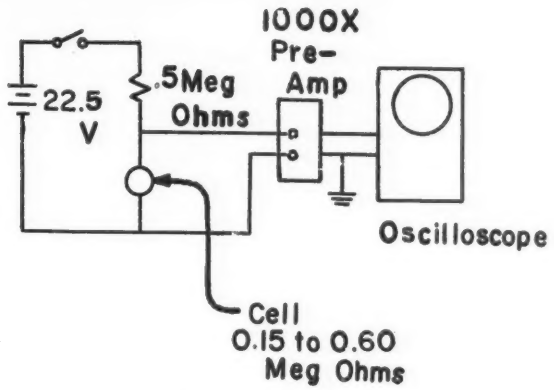



Figure 4. Lead Sulfide Cell Circuit.

APPARATUS

In this study, grinding was of the surface grinding type, and a 6-inch rotary machine was used having infinitely variable spindle, work, and downfeed speeds. A 7-inch diameter cup grinding wheel was used as shown in Figure 3. The small hole ($\sqrt{2}$/16 inch dia.) drilled radially through the grinding wheel, as shown, enabled the lead sulfide cell to sight upon the workpiece surface immediately after grinding. Since the resistance of the lead sulfide cell is sensitive to changes in its ambient temperature as well as to infrared radiation, the cell was kept at 32° F. by use of the ice-water bath. The hole in the sighting tube was masked to provide a square opening 1/16 inch on a side. The cell circuit employed is shown in Figure 4.

The workpiece was a cylindrical specimen of AISI 52100 steel, mounted upon a dynamometer so that the tangential and thrust components of grinding force could be recorded. The grinding wheel had the following characteristics: diameter — 7 inch; abrasive — white aluminum oxide; grain size — 46; hardness — J; structure — 5; bond — vitreous.

The wheel was dressed before each test using a four-sided pyramidal diamond. While dressing, a flat face of the diamond was fed into the wheel .001 inch per pass, with the wheel operating at 2500 fpm. The dressing cross feed was 0.7 fpm. Four successive dressing passes were taken in each case without spark out.

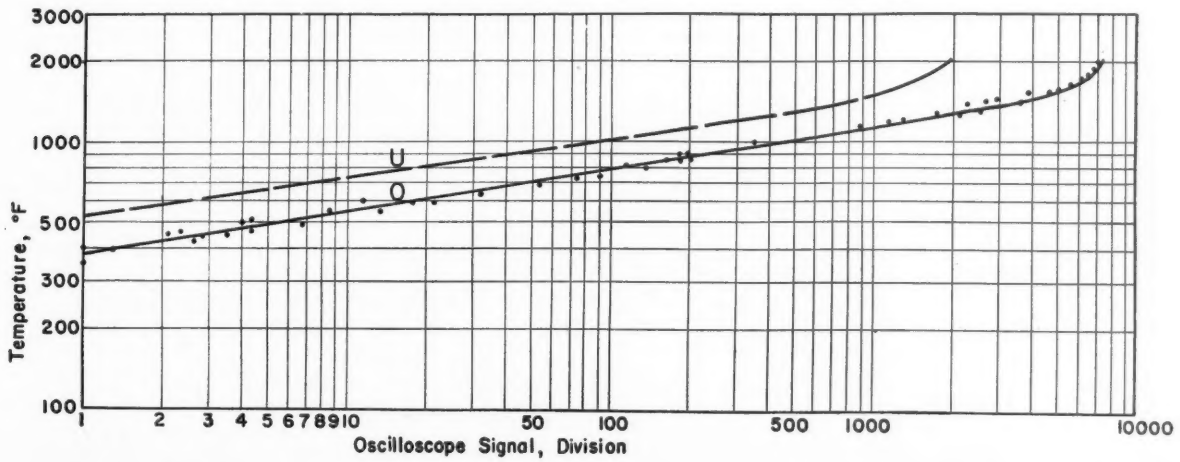



Figure 5. Calibration curve.

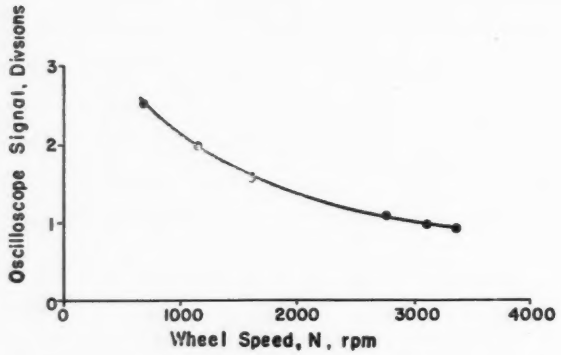



**Figure 6. Variation of relative oscilloscope reading with wheel speed.**

## CALIBRATION

In order to interpret the oscilloscope signal in terms of temperature, it was necessary to calibrate the apparatus. This was done by heating a piece of work material electrically and recording the temperature with a standard alumel-chromel thermocouple attached to the surface. The resulting calibration curve is shown in Figure 5 (curve marked 0 for "oxidized surface"). It is well established that the emissivity of an oxidized steel surface is greater than that for a clean surface or a surface preserved in an inert atmosphere. Relative values of emissivity for clean and oxidized surfaces are available in the literature,[21] and these results were used to convert the oxidized surface calibration curve in Figure 5 to that for an unoxidized surface (marked $U$). Since it is thought that the freshly ground surface sighted by the lead sulfide cell in these experiments will be unoxidized, the $(U)$ curve was used in converting measured values to temperatures.

The calibration curve of Figure 5 was obtained with the grinding wheel operating at 5680 fpm. The frequency response of the cell is such (response time $\cong$ 100 microseconds) that the relative values of oscilloscope readings shown in Figure 6 were obtained for the same temperature when the wheel speed was varied. In practice, the value from this curve should be divided into the actual oscilloscope reading before referring to Figure 5.

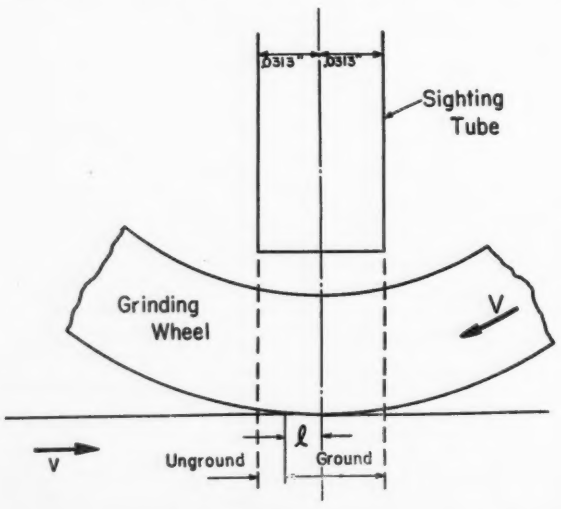



**Figure 7. Relation between sighting tube and ground and unground portions of the surface.**

## INTERPRETATION OF RESULTS

The calibration data of Figures 5 and 6 were obtained by sighting upon radiating surfaces of uniform temperature. However, when the apparatus is used to measure grinding temperatures, the 1/16 × 1/16 inch area looked at is not at uniform temperature for several reasons, and corresponding adjustments should be made in interpreting the data. First of all, only part of the area beneath the citing tube will have been ground. This is illustrated in Figure 7. The radiation from the surface will follow the Stefan-Boltzman Law (radiation flux varies as the fourth power of the absolute temperature of the surface). The unground surface will be at such a low temperature that the radiation from it will be negligible, and it will be as though the unground surface was not present. Experiments in which the sighting area was reduced during calibration revealed that the oscilloscope reading varied directly with the area. Hence, the fact that part of the area is unground may be taken into account simply by dividing the oscilloscope reading by the ratio

$$\left(\frac{\text{citing area ground}}{\text{total citing area}} = \frac{l + .0313}{2(.0313)}\right)$$

where $l$ is the undeformed chip length. Values for $l$ may be computed from the equations of Reference 13. The values of temperature presented in this paper are designated $(\bar{\theta}_c)$, and were obtained by applying the above correction factor to the oscilloscope readings before making use of the calibration curves.

The actual temperature on the plane of shear as a grinding chip is being generated ($\bar{\theta}_s$ in Equation 1) will be considerably higher than the values presented here $(\bar{\theta}_c)$ for the following reasons:

(1) The surface has cooled considerably during the short time it takes the hole in the wheel to become aligned with the sighting tube. During this time no grinding is taking place, and the reading is taken upon a surface that was ground an instant previously.

(2) The surface as ground will not be at uniform temperature, but will be covered by many narrow bands of very high temperature with regions of relatively low temperature between. The method used here measures the total radiation from the surface sighted upon, and not just

**Figure 8. Wheel track in the vicinity of hole in wheel (8X).**

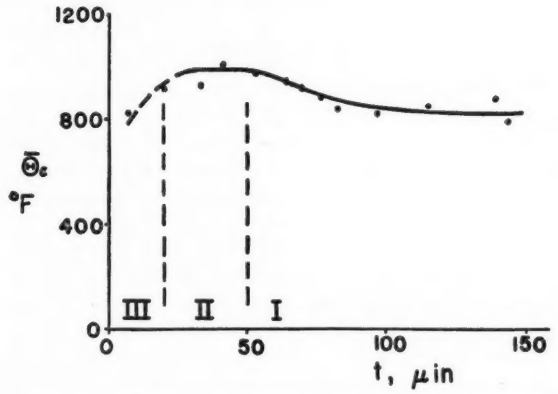



Figure 9. Variation of observed surface temperature ($\bar{\theta}_c$) with undeformed chip thickness ($t$). Wheel speed, 5680 fpm; grinding fluid, none; work material, AISI 52100 steel.

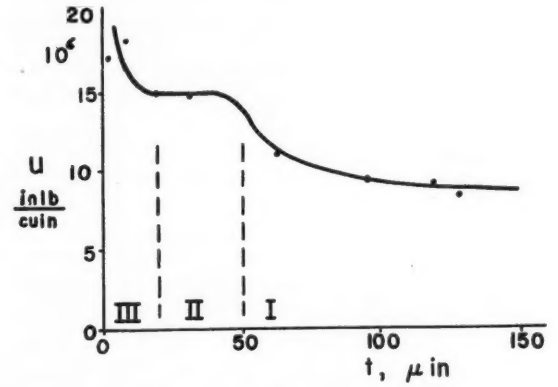



Figure 10. Variation of specific grinding energy ($u$) with undeformed chip thickness ($t$) for conditions of Figure 9.

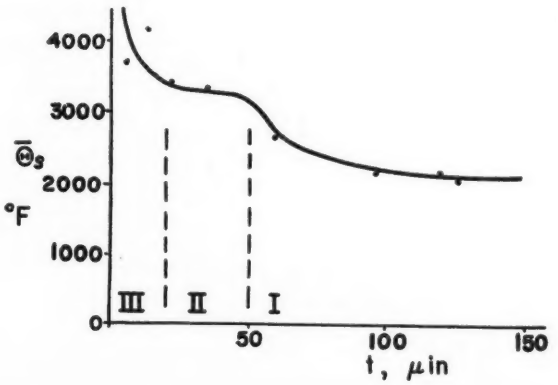



Figure 11. Variation of surface temperature ($\bar{\theta}_s$) computed from data of Figure 10 with undeformed chip thickness ($t$).

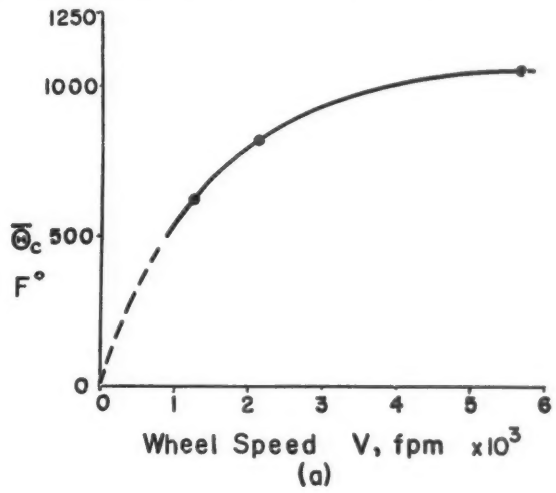
the radiation from the high-temperature bands.

Figure 8 is a picture of the hole in the grinding wheel and the abrasive particles in the vicinity of the hole at a magnification of 8X. This picture was made by rolling the wheel across a glass plate upon which a thin layer of carbon black has been deposited.

It is estimated that the above two effects will cause the mean surface temperature values measured through the hole in the wheel ($\bar{\theta}_o$) to differ from the true maximum values in the scratches ($\bar{\theta}_s$) by a factor of from 2 to 5 (the smaller value for large $t$, the larger for small $t$). While the values of surface temperature presented here are not expected to be in good quantitative agreement with actual peak temperatures, they should be in good qualitative agreement. By studying the effect of different grinding variables upon ($\bar{\theta}_c$), we can gain an insight into their influence upon the actual corresponding peak temperatures.

## DRY GRINDING

A series of tests made at a constant wheel speed of 5680 fpm, but variable values of work speed ($v = 20$ to 260 fpm) and down feed (.0014 to .0102 ipm), yielded the results shown in Figure 9. The values of maximum undeformed chip thickness ($t$) were computed by the method of Reference 13. Here the surface temperature is seen to generally rise in region I and II as the maximum chip thickness ($t$) decreases with a flat portion in the region extending from about $t = 20$ to $t = 50$ microinches. Force readings taken under identical conditions yielded the variation in energy per unit volume ($u$) shown in Figure 10.



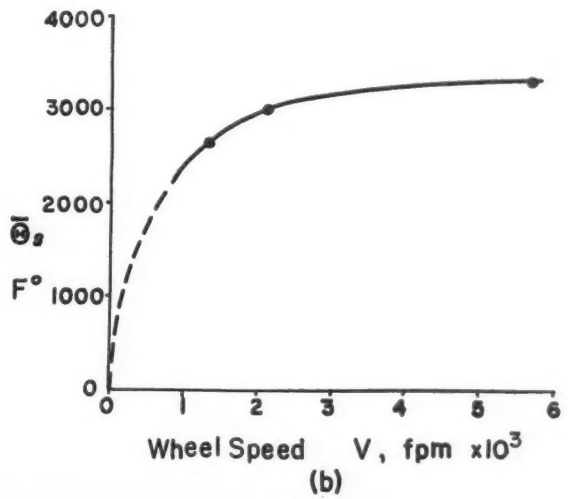




Figure 12. Variation of surface temperatures with wheel speed (a) measured, (b) calculated. Work material, AISI 52100; grinding fluid, none; undeformed chip thickness ($t$), 30 microinches.

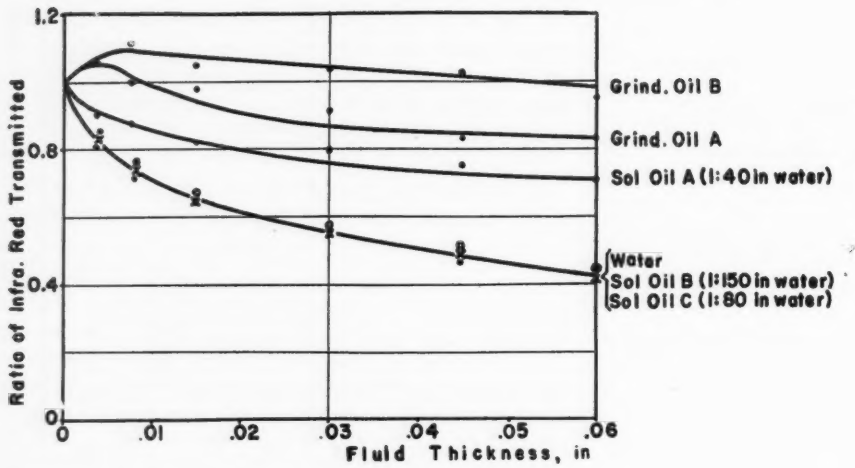



Figure 13. Variation of infrared transmission ratio with fluid thickness.

The shape of this curve is seen to be very similar to that of Figure 9 except in region III, as it should be, since $u$ is by far the most important quantity associated with grinding temperatures (when wheel speed is held constant). The results of Figure 10, when used to compute values of mean shear plane temperature ($\bar{\theta}_s$) by the method of Outwater and Shaw (Equation 2 and Reference 8), gave the values of Figure 11. The shape of this curve is seen to be similar to that of Figure 9 for regions I and II but not for region III. The reason for the lack of agreement in region III lies in the fact that the values of temperature measured ($\bar{\theta}_c$) in this region are lowered an unusual amount by cooling of the surface in the time between grinding and sighting. For regions I and II, the amount of this cooling is about constant and relatively less than in region III.

Comparisons of Figures 9, 10, and 11 reveal:

(1) Good qualitative agreement between the temperature obtained by radiation measurement ($\bar{\theta}_c$) and the actual temperature on the shear plane during grinding ($\bar{\theta}_s$), in regions I and II which are of practical interest.

(2) A ratio of $\bar{\theta}_s$ to $\bar{\theta}_c$ of about 3 as suggested in the section designated "Interpretation of Results."

(3) Verification of the observation that surface temperature varies directly with specific energy ($u$) in dry grinding when the wheel speed is held constant.

Dry grinding tests were also performed at a constant value of $t = 30$ microinch, but variable values of wheel speed ($V = 5680$, 2110, and 1280 fpm), and these results are shown in Figure 12. Here it is again evident that good qualitative agreement exists between the observed (Figure 12a) and calculated (Figure 12b) curves. Both surface temperatures are seen to vary with cutting speed ($V$) approximately as $\sqrt{V}$ just as in the case of ordinary metal cutting. The temperature $\bar{\theta}_s$ given by Equation 2, of course, varies with speed ($V$) as a result of $R$ varying with $V$.

## GRINDING FLUIDS

The infrared absorption characteristics of a variety of grinding fluids were investigated before actual grinding tests were performed. This was done by spacing microscope slides various distances apart with shim stock, and filling the space between the plates with fluid. These cells were then inserted between the grinding wheel and a constant infrared source in line with the sighting tube, and the ratio of infrared transmitted with and without fluid present in the cell determined. Results of a variety of fluids are shown in Figure 13. The oscilloscope readings were actually increased for small thicknesses of some of the fluids, probably due to a change of wave-length as the radiation passed through the fluid.

The abrasive size used in these tests was number 46 (0.015 inch diameter). Thus, the thickness of fluid on the surface sighted upon would have to be considerably less than this value. It is difficult to see how the thickness of the fluid layer could be more than a few thousandths

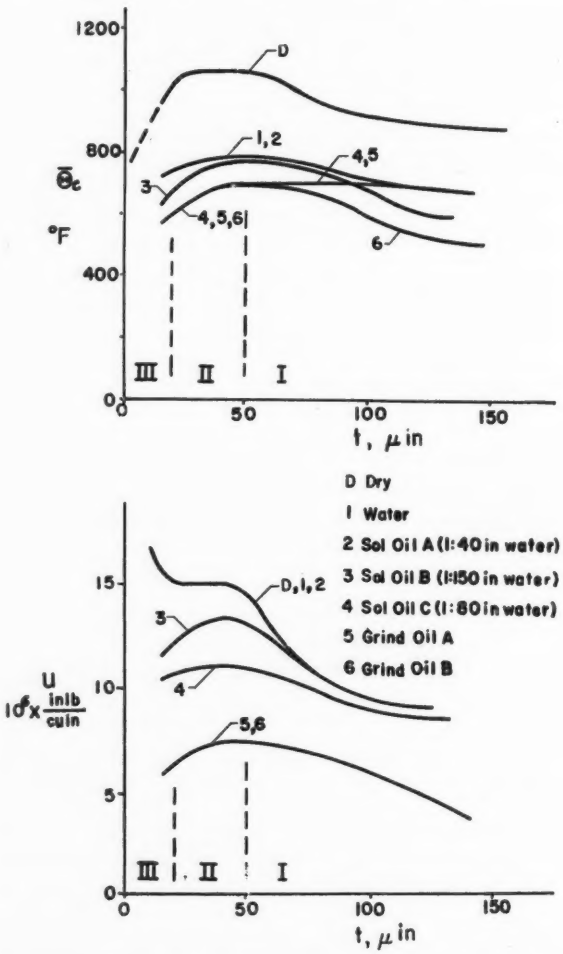



Figure 14. Variation of surface temperature ($\bar{\theta}_c$) and specific grinding energy ($u$) with undeformed chip thickness ($t$) for various grinding fluids. Work material, AISI 52100; wheel speed, 5680 fpm.

of an inch and, since the absorption of such a layer is seen to be small in Figure 13, no correction was made in any of the fluid tests to take care of the presence of a fluid film.

Values of $(\bar{\theta}_c)$ and $(u)$ are given in Figure 14 for a variety of cutting fluids used at a wheel speed of 5680 fpm. While the surface temperature $(\bar{\theta}_c)$ is less in all cases with a fluid than when grinding in air, there is only a slight difference in the temperatures obtained with the several fluids. The grinding oils are seen to give somewhat lower temperatures than the water-base fluids. The action of a grinding fluid is thought to have two components: boundary lubrication, and cooling. The fact that the oil-base materials give somewhat lower temperatures than the water-base fluids suggests that the superior lubrication action of the oils more than offsets their poorer cooling capacity relative to the water-base materials. The fact that both oil- and water-base materials are used in practice is consistent with the observation that there is relatively little difference between the two types of fluids. However, it is evident that in all regions the surface temperature is reduced significantly by the use of some grinding fluid.

Differences in energy per unit volume $(u)$ with cutting fluid are predominantly a result of boundary lubrication action as opposed to cooling. Thus, we see that the oil-base materials, which are naturally the best lubricants, give the lowest values of $u$ in all regions and, particularly, in regions II and III. It would appear that the boundary lubrication action of grinding fluids can be evaluated by observing relative values of $u$ in fine grinding, where even differences for water-base fluids can be observed. It is to be expected that those fluids giving the lowest values of $(u)$ will provide the best finish.

The relative cooling action of fluids can be observed by comparing the $u$ vs $t$, and $\bar{\theta}_c$ vs $t$ curves. For dry grinding at constant wheel speed, $R$ in Equation 2 is constant, and $u$ and $\bar{\theta}_c$ vary with $t$ in the same way. However, when a cutting fluid is used, $R$ is no longer constant, as a result of cooling and, hence, $(u)$ and $(\bar{\theta}_c)$ may vary in different way with $(t)$.

It would appear that the reason all fluids give essentially the same $\bar{\theta}_c$ vs $t$ curve lies in the fact $\bar{\theta}_c$ is a function of both boundary lubrication and cooling, and those fluids that are good lubricants are relatively poor coolants and vice versa.

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See also: *International Critical Tables, 1* and *5*, Nat. Research Council, Washington, D. C.



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**It's Detroit in '57**

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# An Analysis of Orifice- & Capillary-Compensated Hydrostatic Journal Bearings

**Hydrostatic bearings possess several desirable characteristics. These include extremely low friction during starting or during operation at very low or even zero speeds, and an ability to operate on a wide range of fluids, including liquids and gases, as lubricants. A compensated hydrostatic journal bearing is one in which the lubricant is supplied at various points through restrictions in the form of orifices or capillaries which tends to make the shaft seek a central position regardless of the direction of load application. This paper supplies analytical and experimental data of interest to the bearing designer.**

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AUTHORS' NOTE: The bearings discussed in this paper frequently have been referred to as "externally pressurized journal bearings." However, it is felt that this term is not accurately descriptive, since it does not disassociate this bearing category from either the plain journal bearing which may literally be externally pressurized, or the plain hydrostatic bearing which does not utilize any compensating elements such as orifices or capillaries. In the authors' opinion, the terminology "compensated hydrostatic journal bearings" is to be preferred.

## NOMENCLATURE

The following nomenclature is used in the text; some of the terms are shown diagrammatically in Figure 1.

| | |
|---|---|
| $p_1, p_2, p_3, p_4$ | — pressure in pockets 1, 2, 3, 4 respectively, psig |
| $p_s$ | — supply pressure, psig |
| $\Delta p$ | — pressure drop, psig |
| $Q_T$ | — bearing total flow rate, in.$^3$/sec |
| $\phi$ | — load angle, radians |
| $e$ | — displacement or eccentricity, in. |
| $C$ | — radial clearance = brg. rad.-shaft rad., in. |
| $\epsilon$ | — eccentricity ratio = e/$C$, dimensionless |
| $h_o$ | — min. film thickness = $C(1 - \epsilon)$, in. |
| $W$ | — load, lb |
| $R$ | — journal radius, in. |
| $D$ | — journal diameter, in. |
| $L$ | — axial bearing length, in. |
| $L_p$ | — axial pocket length, in. |
| $l$ | — axial pocket land length, in. |
| $l_c$ | — circumferential distance between pockets, in. |
| $L_t$ | — capillary tubing length, in. |
| $d_t$ | — capillary tubing diameter, in. |
| $d_o$ | — orifice diameter, in. |
| $K_o$ | — orifice flow coefficient, dimensionless |
| $\mu$ | — viscosity in reyns, lb-sec/in.$^2$ |
| $\gamma$ | — weight density, lb/in.$^3$ |
| $g$ | — acceleration due to gravity = 386 in./sec$^2$ |
| $\rho$ | — mass density = $\gamma/g$, lb-sec$^2$/in.$^4$ |
| $N$ | — shaft rotational speed, rps |
| $m$ | — pocket aspect ratio = $(L_p/D)(l/l_c)$, dimensionless |
| $\lambda$ | — design parameter for orifice-compensated bearing = $(K_o\, d_o^2 l\mu/C^3R)^2 \cdot (18\pi^2/\rho p_s)$, dimensionless |
| $\delta$ | — design parameter for capillary-compensated bearing = $(3\pi/32)(d_t^4 l/C^3 R L_t)$, dimensionless |
| $\omega$ | — velocity parameter = $24\pi N l L_p \mu/C^2 p_s$, dimensionless |

## INTRODUCTION

Although such desirable traits as low starting friction, ability to carry load with no shaft rotation, and high load capacity attainable regardless of lubricant viscosity have made the compensated hydrostatic journal bearing appear attractive to the designer, its wider application has been limited to some extent by the lack of analytical and experi-

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*Editor's Note:* The Appendix to this paper has not been included because of space limitations; readers wishing to obtain copies of this portion of the paper may do so by contacting the authors

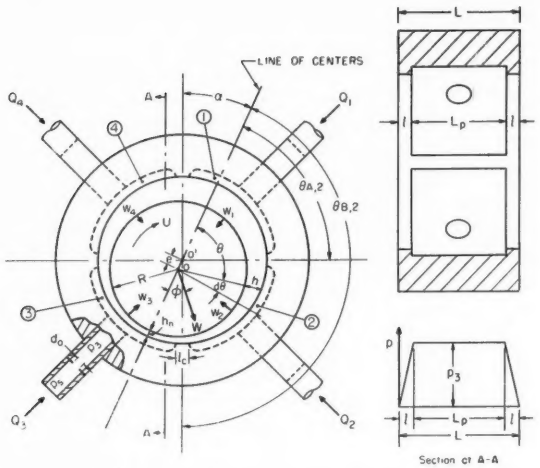


**Figure 1. Nomenclature for 4-pocket orifice-compensated hydrostatic journal bearing.**

mental data. It is the purpose of this paper to provide more information along these lines. While some performance characteristics of a capillary-compensated hydrostatic journal bearing have been presented by Shaw and Macks in their recent book,[1] the characteristics of the orifice-compensated hydrostatic journal bearing have not as yet been evaluated and it is one of the objectives of this paper to present such information. In the course of investigating this particular bearing, certain discrepancies between theory and test were encountered which, upon reassessment of the assumptions on which the theory was based, indicated that the influence of circumferential flow could be appreciable. When the capillary-compensated bearing was tested and analysed in the light of these findings, a similar situation was found to prevail. Therefore, the second objective of this paper is to present a more complete analysis of the capillary-compensated bearing and to supplement this analysis with test data. In addition, the effect of journal rotation on the performance characteristics of this class of bearings is discussed.

## TYPES OF LOADING

Although compensated hydrostatic journal bearings are used frequently as vertical guide bearings where the load may act in any direction, there are many applications where the direction of the load is known. Under these circumstances, one has the option of arranging the pockets in various configurations with respect to the applied load, and

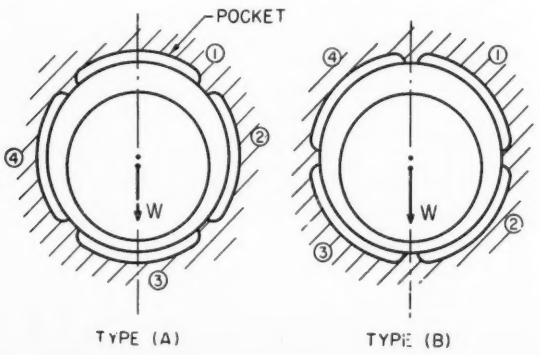


**Figure 2. Types of loading.**

it is not at all necessary that the pockets be of equal size or that they be spaced equally around the bearing circumference. Moreover, the pockets may be cut in the journal rather than in the bearing. However, since the usual practice is to construct the bearing with four identical equally spaced pockets, the present discussion will be limited to bearings of this design. Figure 1 shows the relationship between the journal and the bearing for some arbitrary direction of the load application. Although the load line may assume any orientation with respect to a given pocket, only the two directions of load application shown in Figure 2 as type A and type B will be discussed since they represent the two limiting cases. In type A the load line passes through the center of a pocket, whereas in type B it passes midway between two pockets.

## PRINCIPLE OF OPERATION

The principle of operation of compensated bearings will be briefly reviewed with the aid of Figure 3 for the case of type A loading and no journal rotation. Each of the four pockets shown in the insert in Figure 3 is connected to a source of lubricant which is supplied at a constant pressure. However, in order to reach a pocket, the lubricant must first pass through a hydraulic resistance whose magnitude is usually made the same value for each line. These resistances may take the form of either orifices or capillary tubes, whence comes the designation: orifice-compensated or capillary-compensated bearing. The fluid from each pocket escapes in the axial direction out both ends of the bearing through the clearance space between the bearing and the journal, (i.e., across lands $l$ in Figure 1). In addition there may be circumferential flow from pocket to pocket, but this does not alter the main principle of operation; its effect will be discussed later. When the journal and bearing centers are coincident (that is, $e = 0$) as indicated by point 0′ in Figure 3, the areas of escape are the same for each pocket and, as a result, the pressure in each pocket will be identical. However, when the journal is displaced by an amount $e$, under the action of the load $W$, the area of escape for pocket 3 decreases, restricting the flow and causing the pocket pressure $p_3$ to build up to the

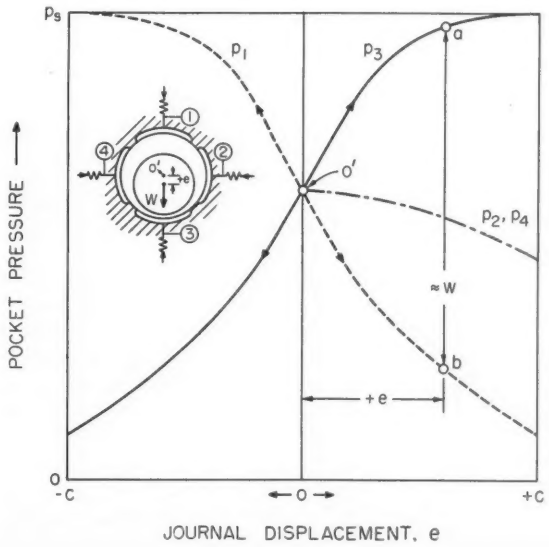


**Figure 3. Typical pocket pressure vs. journal displacement curves for compensated hydrostatic bearings.**

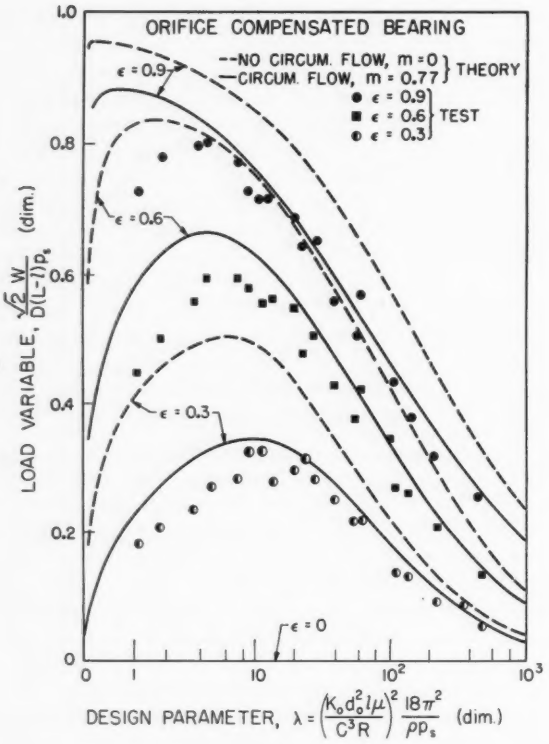


**Figure 4. Load variable vs. design parameter for orifice-compensated hydrostatic journal bearing (Type A loading, no rotation).**

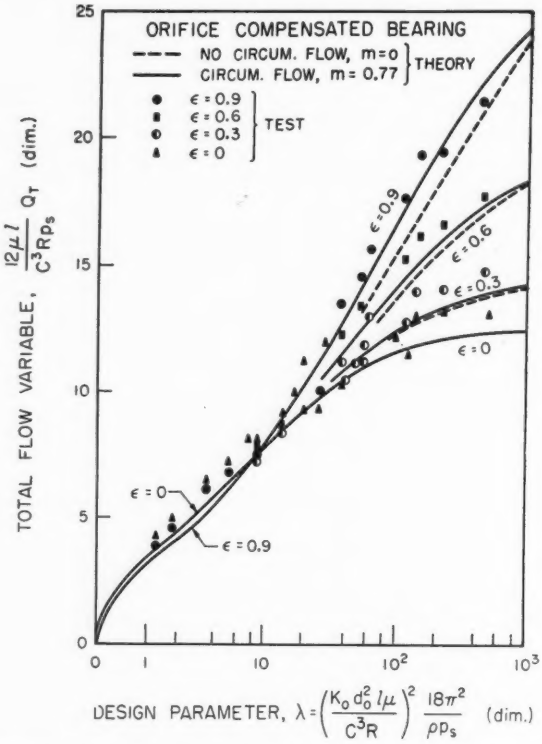


**Figure 5. Total flow variable vs. design parameter for orifice-compensated hydrostatic journal bearing (Type A loading, no rotation).**

value indicated by point *a* on the solid curve in Figure 3. On the other hand, the escape area of pocket 1 is increased allowing more fluid to escape. This increased flow through the hydraulic resistance, however, causes the pocket pressure $p_1$ to drop to the value indicated by point *b* in Figure 3. The pressure differential existing between pockets 3 and 1 produces a force tending to restore the journal to a concentric position within the bearing. The magnitude of this force is proportional to the vertical distance between points *a* and *b*. It becomes evident from Figure 3 that as the journal displacement *e* is increased, the restoring force increases, reaching a maximum value where the journal is displaced an amount equal to the radial clearance. By proper selection of the design parameters, the journal can be made to come to rest at a given displacement *e*, at which point the restoring force and the load *W* will be in equilibrium. In practice, the designer will limit the displacement *e* in order to ensure a safe value of the minimum film thickness.

Because of the symmetrical location of pockets 2 and 4 with respect to the line of centers, the pressure in each of these pockets will be identical. Therefore, there will be no force normal to the displacement, and the resultant or restoring force will be co-linear with the line of centers. (This co-linearity will exist for type B loading as well as for type A for which Figure 3 is drawn.) Since the escape areas of pockets 2 and 4 are not greatly influenced by a vertical movement of the journal, these pocket pressures which are indicated by the broken curve in Figure 3 do not vary appreciably with the displacement *e*.

When rotation takes place, the basic principle of operation of this bearing remains essentially unchanged. However, one of the principal features of this type of bearing is that, unlike the plain journal bearing, rotation is not necessary for developing load capacity.

## THEORETICAL ANALYSES

The various analyses that were made of both the orifice- and capillary-compensated bearings and the assumptions on which they were based may be found in detail in the appendix. Basically, two theories were employed in the analyses made of both the orifice- and the capillary-compensated bearings, namely: (a) that no circumferential flow from pocket to pocket took place ("no circumferential flow theory"), and (b) that circumferential flow did occur ("circumferential flow theory"). The effects of including circumferential flow in the analyses will be discussed shortly.

### Design Parameters

Typical performance curves resulting from the analysis for the orifice-compensated bearing with type A loading are shown in Figures 4 and 5. The test points in these two figures (and Figures 6 and 7) should be disregarded temporarily as their discussion is reserved for the section dealing with tests.

Figure 4 shows the load variable, $\sqrt{2}W/D(L-l)p_s$, which for given values of $D$, $L$, $l$ and $p_s$ is directly proportional to the load capacity $W$, plotted against the "design parameter" for orifice compensated bearings $\lambda$, where

$$\lambda = \left(\frac{K_o d_o^2 l\mu}{C^3 R}\right)^2 \frac{18\pi^2}{\rho p_s} \qquad [1]$$

and is dimensionless. The symbol $\lambda$ is designated as being a "design parameter" for orifice-compensated bearings because it contains such quantities as the orifice diameter $d_o$, the radial clearance $C$, the supply pressure $p_s$, and other quantities which the designer may vary at will to attain an optimum design. The curves in Figure 4 are drawn for various values of the eccentricity ratio $\epsilon$ which is a measure of the journal position within the bearing. The minimum film thickness is related to the eccentricity ratio through the well-known expression

$$h_0 = C(1 - \epsilon) \qquad [2]$$

Figure 4 shows that an optimum value of the load variable exists for a fixed value of the eccentricity ratio. These optimum points are indicated by the peaks of the curves which are seen to occur at definite values of the design parameter.

Figure 5 is a plot of the total flow variable, $12\mu l Q_T/C^3 R p_s$, which for fixed values of $\mu$, $l$, $C$, $R$ and $p_s$ is directly proportional to the total required bearing flow $Q_T$. The use of Figures 4 and 5 will be illustrated later by a typical design problem.

Figures 6 and 7 resulting from the analysis for the capillary-compensated bearing show the load variable and total flow variable plotted for various eccentricity ratios against a dimensionless quantity $\delta$, where

$$\delta = \frac{3\pi}{32}\left(\frac{d_t^4 l}{C^3 R L_t}\right) \qquad [3]$$

and is called the "design parameter" for capillary-compensated bearings since it contains the capillary tubing diameter $d_t$, tubing length $L_t$, and other quantities over which the designer may exercise some control. It is evident that by proper choice of $\delta$, optimum load capacity for a capillary-compensated bearing may also be obtained.

A comparison of the maximum values of the load variable shown in Figures 4 and 6 indicates that for a given eccentricity ratio the orifice-compensated bearing has more load capacity than the capillary-compensated bearing. This is illustrated in Table I. Another interesting comparison is that the load capacity of an orifice-compensated bearing is affected by changes in the lubricant viscosity since the quantity $\mu$ appears in the design parameter $\lambda$. Since this term is lacking in the design parameter $\delta$, the load capacity of a capillary-compensated bearing is independent of the lubricant viscosity (provided the journal does not rotate).

**Table I. Illustrating superior load capacity of orifice-compensated bearing (no rotation).**

| $\epsilon$ | MAX. LOAD CAPACITY, CAPILLARY COMP. BRG. / MAX. LOAD CAPACITY, ORIFICE COMP. BRG. | | |
|---|---|---|---|
| | NO CIRCUM. FLOW THEORY, m = 0 | CIRCUM. FLOW THEORY, m = 0.77 | EXPERIMENTAL, m = 0.77 |
| 0.3 | 0.78 | 0.81 | 0.79 |
| 0.6 | 0.85 | 0.83 | 0.81 |
| 0.9 | 0.96 | 0.89 | 0.90 |

### Circumferential Flow

Since the pressure in the various pockets of a com-

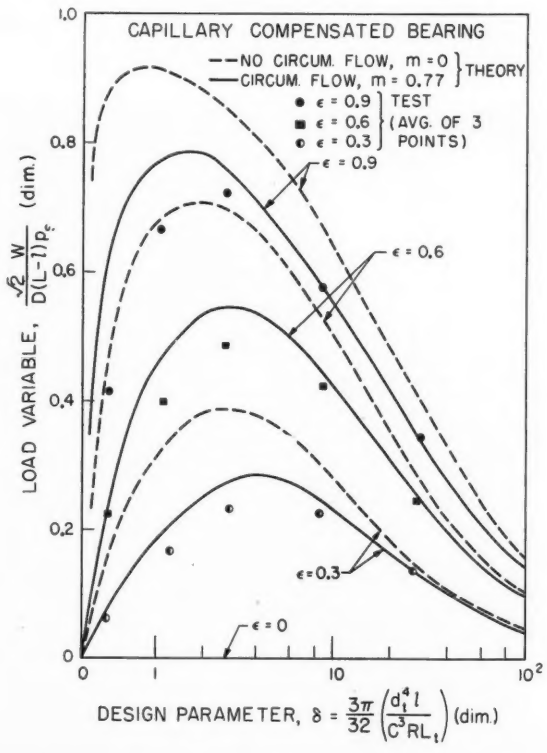


**Figure 6. Load variable vs. design parameter for capillary-compensated hydrostatic journal bearing (Type A loading, no rotation).**

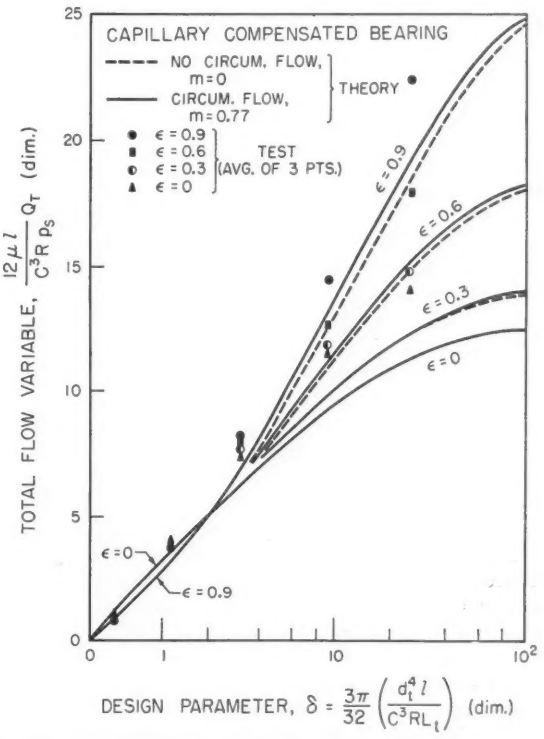


**Figure 7. Total flow variable vs. design parameter for capillary-compensated hydrostatic journal bearing (Type A, no rotation).**

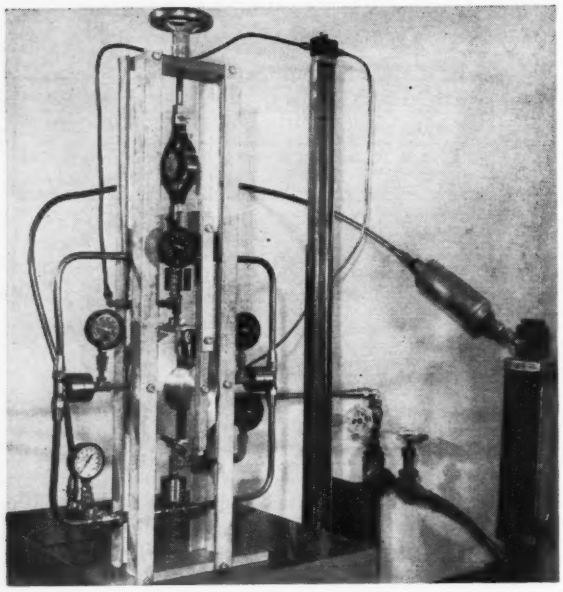

**Figure 8. Test apparatus.**

pensated hydrostatic bearing may differ appreciably as shown in Figure 3, pressure-induced flow may take place in the circumferential direction from pocket to pocket (see Figure 16) as well as axially out both ends of the bearing. The amount of flow which escapes circumferentially from any given pocket into an adjacent pocket is related directly to the pocket and bearing dimensions, and is reflected by the magnitude of the "pocket aspect ratio" $m$, where

$$m = \left(\frac{L_p}{D}\right)\left(\frac{l}{l_r}\right) \qquad [4]$$

and is dimensionless. The typical manner in which the pocket aspect ratio influences the pocket pressures is illustrated in Figure 11 where the pocket pressure ratios (which for a constant supply pressure $p_s$ are directly proportional to the pocket pressures) are plotted against the pocket aspect ratio $m$ for an orifice-compensated bearing with type A loading operating with $\lambda = 5$ and $\epsilon = 0.6$ and no rotation. When $m = 0$, there is no circumferential flow, and those pressures exist in the various pockets as calculated on the basis that all the lubricant escapes in the axial direction which corresponds to the "no circumferential flow theory." The load capacity of the bearing is proportional to the pressure differential existing between the top and bottom pockets, that is, proportional to $(p_3/p_s - p_1/p_s)$. As $m$ is increased, more fluid flows circumferentially out of pocket 3 into pockets 2 and 4, depleting the pressure in pocket 3. (As is evident from Eq. [4], this increase in $m$ can be achieved by either decreasing the circumferential distance between the pockets, increasing the axial land length $l$, increasing the pocket axial length $Lp$, or decreasing the bearing diameter $D$). Since the journal as shown in Figure 11 is displaced vertically downward ($\epsilon = 0.6$), more fluid escapes circumferentially from pockets 2 and 4 into pocket 1 then enters pockets 2 and 4 from pocket 3. The net effect is a lowering of the pressures in pockets 2, 3 and 4, and an increase in the pressure in pocket 1, as indicated by pressure curves in Figure 11. As the pocket aspect ratio $m$ is increased still further, the curves for pocket 3 and pocket 1 continue to approach each other, thus diminishing the load capacity of the bearing.

Comparison of the dotted and solid curves in Figures 4 and 6 shows the large loss in load capacity that may be experienced over a range of the design parameter when $m = 0.77$. The extent that load capacity may be decreased by increasing the pocket aspect ratio $m$ is shown in Figure 12 for both an orifice- and capillary-compensated bearing. The values of the design parameters $\lambda$ and $\delta$ for which this figure was drawn were taken to be near the optimum load values indicated in Figures 4 an 6 respectively for an eccentricity ratio, $\epsilon = 0.6$, and does not necessarily present the worst case. It is obvious that if the pocket aspect ratio is made excessively large, the load capacity of the bearing may be reduced considerably. The dashed curve in Figure 12, being lower than the solid curve, indicates that the capillary-compensated bearing (with type A loading) may be more susceptible to circumferential flow effects than the orifice-compensated bearing (with type A loading). Comparison of the broken curves reveals that bearings with type B loading undergo a greater loss in load capacity than those with type A loading because of circumferential flow effects. Figure 12 also shows that the total flow is affected by the pocket aspect ratio, but not to the same extent as load capacity.

### Type of Loading

The effect of type of loading has been partly illustrated in Figure 12 for one value of the design parameter $\delta$ for a capillary-compensated bearing. Figure 13 shows the influence of the type of loading on a capillary-compensated bearing over a wide range of the design parameter. When circumferential flow is neglected (i.e., $m = 0$), there is no appreciable difference in the optimum load capacity, as can be seen by comparing the ordinate values of the peaks of the dotted curves. The dotted curves also illus-

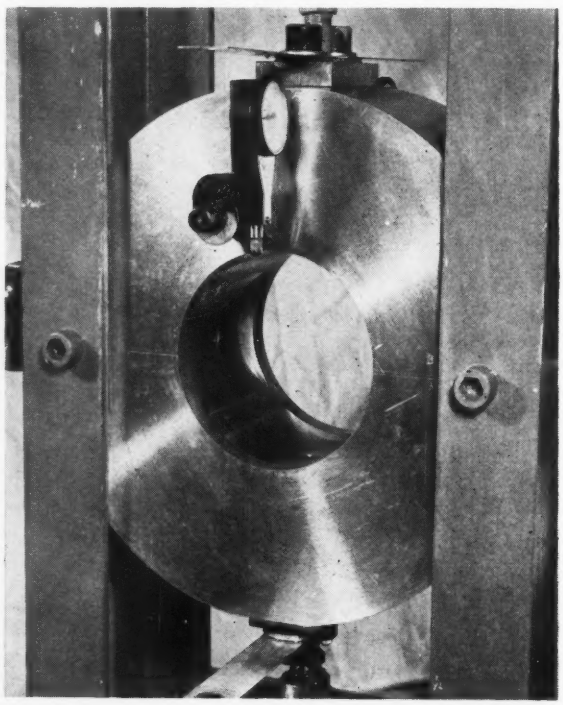

**Figure 9. Test bearing.**

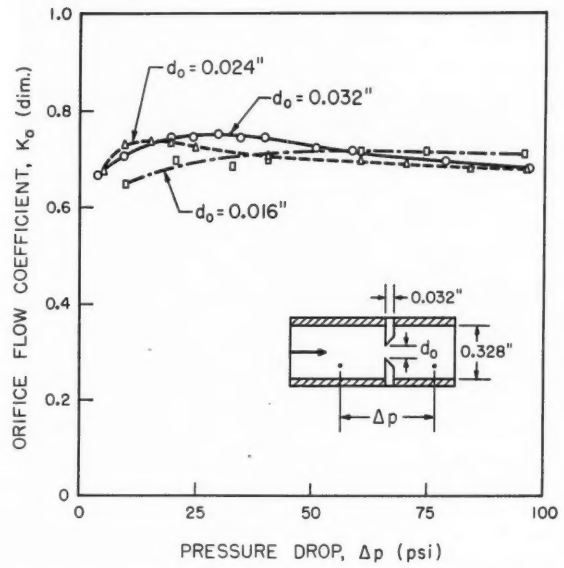


**Figure 10. Flow coefficient vs. pressure drop curves for test orifices.**

trate that a decrease in load capacity is obtained with type B loading in the low range of the design parameter, although this is offset to some degree by a slight increase in load capacity when $\delta$ is greater than 3. When the effect of circumferential flow is accounted for, however, as illustrated by the solid curves drawn for $m = 0.77$, not only is the optimum load capacity decreased with type B loading, but the load capacity is diminished over a longer range of the design parameter. These differences would probably be accentuated if a more pessimistic value of the pocket aspect ratio were chosen. Figure 13 indicates that designs based on type B loading would be the more conservative.

## TESTS

Tests were conducted on the four-pocket aluminum bearing arranged for type A loading and no rotation as shown in Figures 8 and 9. Pertinent bearing dimensions were: $L = D = 3.25''$, $L_p = 2.50''$, $l = l_c = 0.375''$. Substituting these values in Eq. [4] gives a pocket aspect ratio for the test bearing of $m = 0.77$. Two aluminum journals were used giving radial clearances of 0.0024" and 0.0034". These clearances were obtained by separate journal and bearing measurements, and checked by measuring the total play when the journal was positioned in the bearing. Since it was desired to have only one bearing in the test apparatus and, since a single hydrostatic bearing cannot compensate for axial misalignment, it was necessary to prevent the journal from becoming misaligned during the tests. As no rotation was involved, this was done by employing axial flexure strips (see Figure 9). These strips were fastened to vertical bars which were bolted to each end of the journal. Load was applied to the journal by means of a system of links connected to the handwheel shown in Figure 8 and measured with a proving ring. The journal position was measured by means of dial indicators attached to each end of the bearing. In all tests, a light turbine oil having a viscosity of $4.3 \times 10^{-6}$ reyns at 100° F. and $0.68 \times 10^{-6}$ reyns at 210° F. was used as the lubricant. The oil was filtered and supplied at pressures of 30, 60 and 90 psig by means of a gear pump. Pocket pressures were measured with calibrated pressure gages; flow measurements made with calibrated flowmeters. The inlet oil

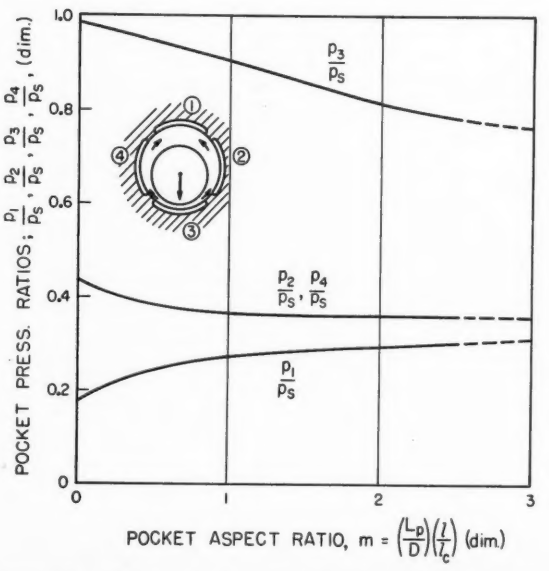


**Figure 11. Effect of pocket aspect ratio on pocket pressures (Orifice-compensated bearing, Type A loading, $\lambda = 5$, $\epsilon = 0.6$, no rotation).**

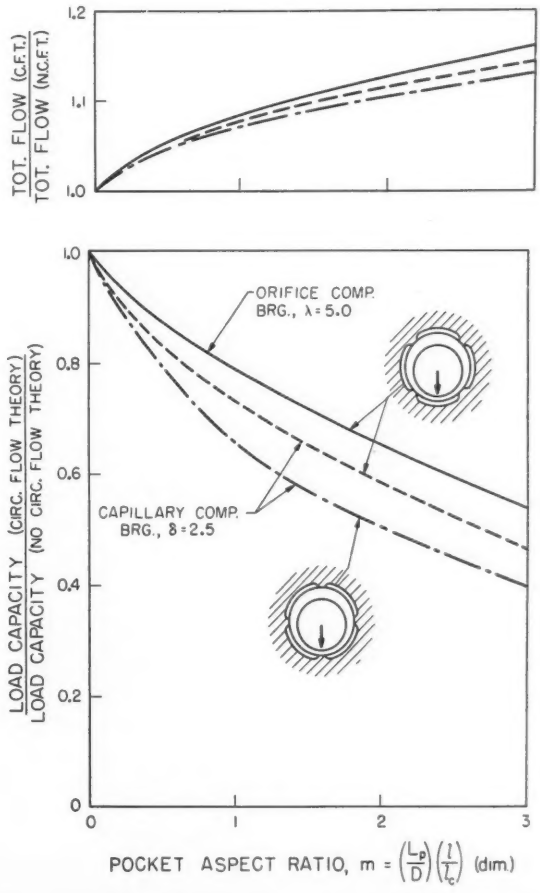
$\frac{\text{TOT. FLOW (C.F.T.)}}{\text{TOT. FLOW (NC.F.T.)}}$
$\frac{\text{LOAD CAPACITY (CIRC. FLOW THEORY)}}{\text{LOAD CAPACITY (NO CIRC. FLOW THEORY)}}$
ORIFICE COMP. BRG., λ = 5.0
CAPILLARY COMP. BRG., δ = 2.5
POCKET ASPECT RATIO, $m = \left(\frac{L_p}{D}\right)\left(\frac{l}{l_c}\right)$ (dim.)

**Figure 12. Effect of pocket aspect ratio on load capacity and total flow (Orifice- and capillary-compensated bearings, $\epsilon = 0.6$, no rotation).**

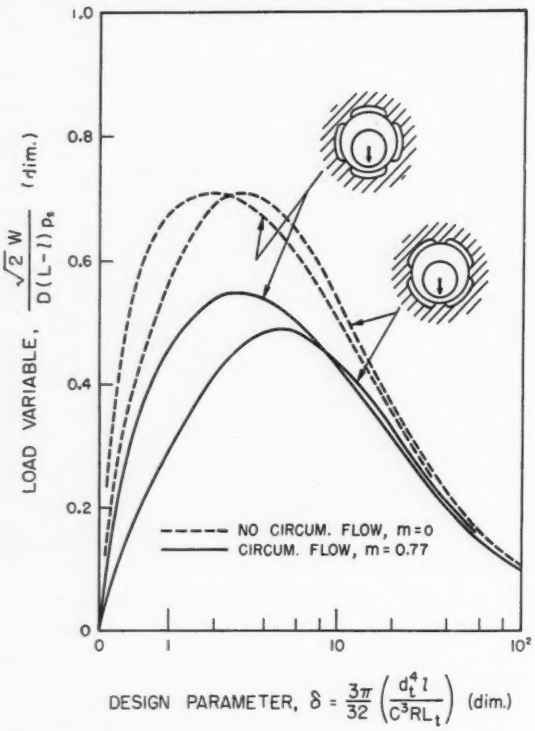
LOAD VARIABLE, $\frac{\sqrt{2}\,W}{D(L-l)p_s}$ (dim.)
NO CIRCUM. FLOW, m=0
CIRCUM. FLOW, m=0.77
DESIGN PARAMETER, $\delta = \frac{3\pi}{32}\left(\frac{d_t^4 l}{C^3 R L_t}\right)$ (dim.)

**Figure 13. Effect of type of loading on load capacity (Capillary-compensated bearing, $\epsilon = 0.6$, no rotation).**

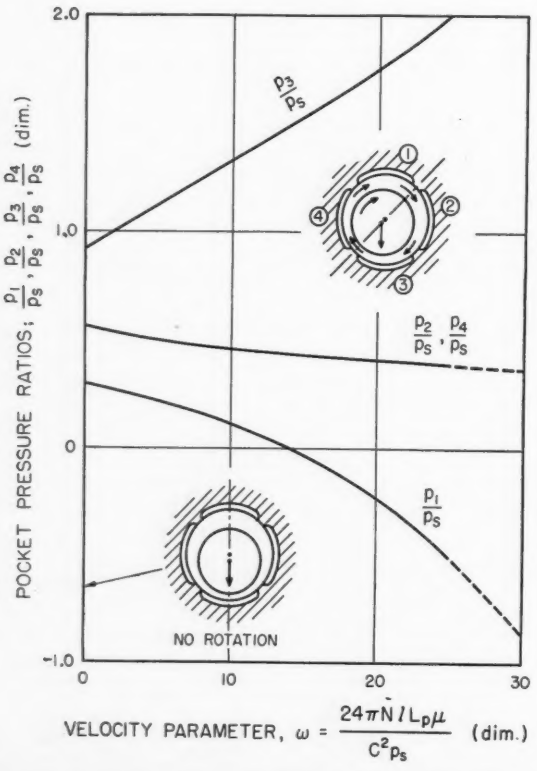
POCKET PRESSURE RATIOS; $\frac{p_1}{p_s}, \frac{p_2}{p_s}, \frac{p_3}{p_s}, \frac{p_4}{p_s}$ (dim.)
$\frac{p_3}{p_s}$
$\frac{p_2}{p_s}, \frac{p_4}{p_s}$
$\frac{p_1}{p_s}$
NO ROTATION
VELOCITY PARAMETER, $\omega = \frac{24\pi N l / L_p \mu}{C^2 p_s}$ (dim.)

**Figure 14. Theoretical effect of rotation on pocket pressure ratios (Capillary-compensated bearing, Type A loading, $\delta = 5$, $\epsilon = 0.6$).**

temperature was recorded, and was found to be very nearly equal to the temperature of the oil escaping from the bearing which was about 80° F. Tests were conducted using both orifices and capillaries as the compensating elements. Sharp-edged orifices were made from 1/32″ brass sheet, and measured with a microscope. The average diameter of any one set of orifices was found to vary less than 3.5 per cent. Each orifice was calibrated in position, and typical orifice flow coefficient curves for the three orifices used in the tests are shown in Figure 10. For plotting the test data, an orifice coefficient of $K_o = 0.7$ was used, since the calibration curves were reasonably flat at this value over the range of pressure drops used in the tests. Various lengths of one-eighth inch O.D. soft copper tubing ranging from 60″ down to 2-1/8″ were used for the capillary-compensated bearing tests. The average diameter of each length of tubing was found by measuring the flow through the tubing while it was installed on the test apparatus, and calculating the diameter using Poiseuille's equation. It was found that the various lengths yielded reasonably constant diameters, provided they were all cut from the same coil of tubing.

## COMPARISON OF THEORY & TESTS

A comparison of the dotted and solid curves in Figures 4 and 6 shows that there exists an appreciable discrepancy in load capacity as predicted by the two theories. This is especially true for eccentricity ratios of $\epsilon = 0.3$ and

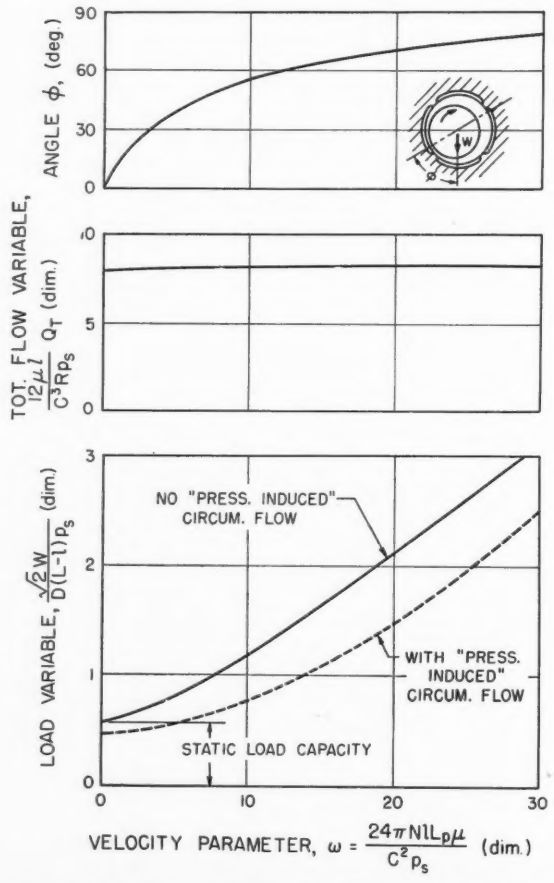
ANGLE $\phi$, (deg.)
TOT. FLOW VARIABLE, $\frac{12\mu l}{C^3 R p_s} Q_T$ (dim.)
LOAD VARIABLE, $\frac{\sqrt{2}\,W}{D(L-l)p_s}$ (dim.)
NO "PRESS. INDUCED" CIRCUM. FLOW
WITH "PRESS. INDUCED" CIRCUM. FLOW
STATIC LOAD CAPACITY
VELOCITY PARAMETER, $\omega = \frac{24\pi N l L_p \mu}{C^2 p_s}$ (dim.)

**Figure 15. Theoretical effect of rotation on journal position, total flow, and load capacity (Capillary-compensated bearing, Type A loading, $\delta = 5$, $\epsilon = 0.6$).**

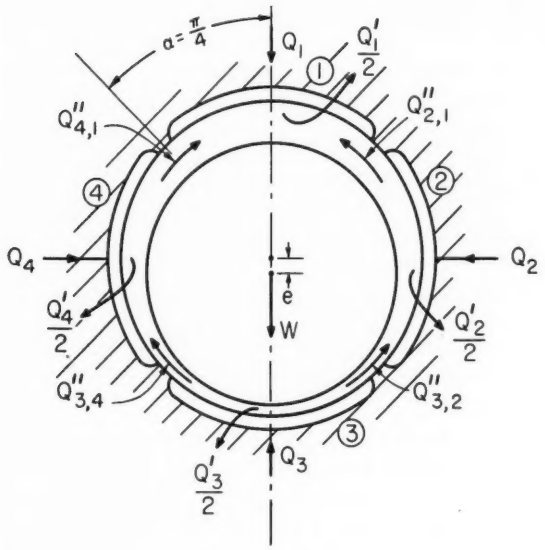


Figure 16. Compensated bearing with circumferential flow (Type A loading, no rotation).

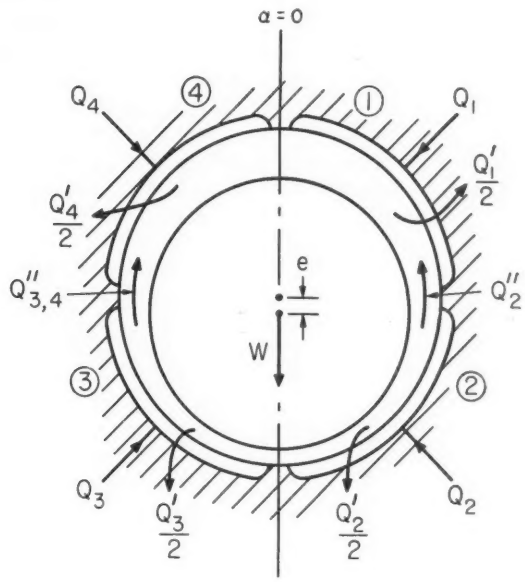


Figure 17. Compensated bearing with circumferential flow (Type B loading, no rotation).

$\epsilon = 0.6$. The test points are in much better agreement with the "circumferential flow theory" than with the "no circumferential flow theory" over the entire design parameter range, and substantiate the importance of the effect of circumferential flow. At a relatively high eccentricity ratio of $\epsilon = 0.9$, although the better correlation still exists with respect to the circumferential flow theory, the test points are appreciably lower than the theoretical solid curves for low values of the design parameter. This deviation is more marked for the orifice-compensated bearing, and may possibly be partially attributed to a decrease in the flow coefficient for the orifice in the bottom pocket resulting from the very low pressure drop which may prevail as a result of the relatively high eccentricity ratio. Another explanation may be that at high eccentricity ratios, the relative curvature of the circumferential land $l_c$ and the journal is accentuated in the region of the bottom pocket, and that the approximation (made in the appendix) that circumferential flow from this pocket may be calculated on the basis of flow through parallel plates becomes worse.

Figures 5 and 7 illustrate that the total flow is increased by accounting for the effect of circumferential flow, although the increase is negligible especially at low eccentricity ratios. With few exceptions the test points seem to be consistently higher than the theoretical values, especially in Figure 7. This may be attributed in part to experimental errors in measuring the flow, bearing clearances, and capillary diameters, but it is felt that it may also be significant in that it exhibits a qualitative as well as a quantitative agreement in favor of the circumferential flow theory.

In general, Figures 4, 5, 6 and 7 illustrate excellent qualitative agreement with regard to both load capacity and flow. Best quantitative agreement occurs, however, with the solid curves which account for circumferential flow effects in the test bearing. That this is true for the capillary-compensated bearing, as well as for the orifice-compensated bearing, appears to be at variance with the capillary-compensated bearing tests of Shaw and Macks who obtained very good agreement with their analytical treatment which neglected the effects of circumferential flow. An explanation on the basis of negligible circumferential flow effects may be given, provided the pocket aspect ratio of their test bearing was relatively low. However, since no pertinent bearing dimensions other than the diameter was given, it was not possible to calculate an overall pocket aspect ratio.

## THEORETICAL EFFECT OF ROTATION

In order to determine the effect of journal rotation on compensated bearings, an approximate theoretical analysis has been made in Section 7 (Analysis IV) of the appendix. The principal assumption in this analysis is that the lubricant will be pumped from pocket to pocket in the circumferential direction because of the shearing action of the journal. This "velocity-induced" circumferential flow is

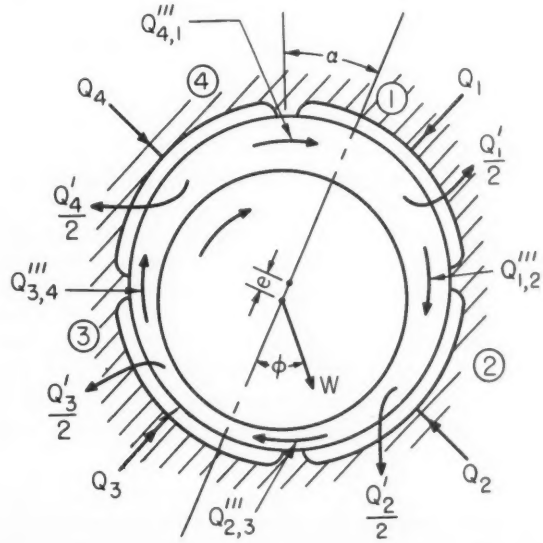


Figure 18. Compensated bearing with velocity-induced circumferential flow (Arbitrary loading, rotation).

not to be confused with the "pressure-induced" circumferential flow discussed previously, and which also exists when rotation takes place. However, in order to simplify the analysis and to investigate the influence of rotation separately, the pressure-induced circumferential flows have been neglected. The effect of rotation is reflected by the magnitude of the dimensionless "velocity parameter", $\omega$, given as

$$\omega = \frac{24\pi N l L_p \mu}{C^2 p_s} \qquad [5]$$

where $N$ is the speed of rotation. Eq. [5] shows that when the journal is stationary, that is, $N = 0$, the velocity parameter $\omega$ is zero, and the pocket pressures which prevail are those which can be calculated on the basis of static conditions, that is, no rotation.

The influence of the velocity parameter on the pocket pressure ratios of a capillary-compensated bearing with type A loading and operating with an eccentricity ratio, $\epsilon = 0.6$, and a design parameter, $\delta = 5.0$, is illustrated in Figure 14. For a constant supply pressure $p_s$, the pocket pressure ratio is directly proportional to the pocket pressure. Figure 14 shows that as $\omega$ increases, the pressure in pocket 3 increases and is about doubled for this bearing when $\omega = 24$. On the other hand, the pressure in pocket 1 is decreased and may even become negative when $\omega$ is relatively high. This may be interpreted to mean that the journal is pumping lubricant out of pocket 1 at a faster rate than it can be supplied by external means. In an actual bearing, there is a limit to which the pressure may become negative, namely, atmospheric pressure. The pressures in pockets 2 and 4 are necessarily identical, since the figure is drawn for type A loading but decrease slightly with increase in $\omega$. Since the load capacity of this bearing is proportional to the pressure difference existing between pockets 1 and 3, that is proportional to $(p_3/p_s - p_1/p_s)$, and as the pressure curves for $p_3/p_s$ and $p_1/p_s$ diverge as $\omega$ increases, the net effect of journal rotation will be to increase the load capacity of the bearing over the value which it had at static conditions (i.e., at $\omega = 0$). The amount of this increase may be appreciable, and is shown by the solid curve in Figure 15 where the load variable is plotted against the velocity parameter. If the effect of "pressure induced" circumferential flow had been included in this analysis, the net result would be to displace the curve in Figure 15 downward as indicated by the sketched broken curve. The authors intend to check experimentally the effect of rotation on load capacity.

Figure 15 also illustrates that the total flow variable is relatively unaffected by rotation. The manner in which the line of centers precesses in the direction of motion is indicated by the angle $\phi$ in Figure 15.

## DESIGN EXAMPLE

The use of Figures 4 and 5 for the design of an orifice-compensated hydrostatic journal bearing will be illustrated by a typical problem.

Problem: If the following quantities are given for a four-pocket bearing, calculate (a) the minimum film thickness, (b) the pocket aspect ratio, (c) the maximum load capacity, (d) the diameter of the orifices, and (e) the total bearing flow required.

Given quantities:

$\epsilon = 0.6$ $\quad p_s = 100$ psig $\quad D = 3''$
$L = 3$ $\quad l = l_c = 0.30$ $\quad C = 3 \times 10^{-3}$
$K_o = 0.70$ $\quad \gamma = 0.03$ lb/in.$^3$ (giving $\rho = 7.77 \times 10^{-5}$ lb-sec$^2$/in.$^4$)
$\mu = 5 \times 10^{-6}$ reyns (SAE 10 oil at 105F)

Solution:

(a). Using eq. [2] of the text the minimum film thickness is calculated to be

$$h_o = C(1 - \epsilon) = 3 \times 10^{-3}\ (1 - 0.6) = 1.2 \times 10^{-3} \text{ in.}$$

(b). From eq. [4] of the text, the pocket aspect ratio is calculated, giving

$$m = \left(\frac{L_p}{D}\right)\left(\frac{l}{l_c}\right) = \left(\frac{3 - 2 \times 0.30}{3}\right)\left(\frac{0.30}{0.30}\right)$$

$$= 0.80$$

Since this value is close to $m = 0.77$, the solid curves of Figures 4 and 5 may be used.

(c). Referring to Figure 4 it can be seen that the maximum value of the load variable for an eccentricity ratio of $\epsilon = 0.6$ occurs when $\lambda = 4$ (approximately). The load variable for $\lambda = 4$ is

$$\frac{\sqrt{2}\,W}{D(L - l)p_s} = 0.67$$

which gives

$$W = \frac{0.67}{\sqrt{2}} \times D(L - l)p_s$$

$$= \frac{0.67 \times 3(3 - 0.30) \times 100}{\sqrt{2}} = 383 \text{ lb}$$

for the maximum load capacity.

(d). The design parameter $\lambda$ was found from (c) to be

$$\lambda = \left(\frac{K_o d_o^2 l \mu}{C^3 R}\right)^2 \frac{18\pi^2}{\rho p_s} = 4.0$$

which upon solving for $d_o$ gives

$$d_o = \left[\left(\frac{C^3 R}{K_o l \mu}\right)\left(\frac{4.0 \rho p_s}{18\pi^2}\right)^{1/2}\right]^{1/2}$$

Upon substituting the proper values in the above equation, the required orifice diameter is calculated to be

$$d_o = \left[\left(\frac{(3 \times 10^{-3})^3 \times 1.5}{0.7 \times 0.30 \times 5 \times 10^{-6}}\right) \cdot \left(\frac{4.0 \times 7.77 \times 10^{-5} \times 100}{18\pi^2}\right)^{1/2}\right]^{1/2}$$

$$= 0.0225 \text{ in.}$$

(e). Entering Figure 5 with $\lambda = 4$ gives the total flow variable as

$$\frac{12\mu l}{C^3 R p_s} Q_T = 5.5$$

from which the total flow is computed giving

$$Q_T = 5.5 \frac{C^3 R p_s}{12\mu l}$$

$$= \frac{5.5 \times (3 \times 10^{-3})^3 \times 1.5 \times 100}{12 \times 5 \times 10^{-6} \times 0.30}$$

$$= 1.23 \text{ in.}^3/\text{sec}$$

Figures 6 and 7 for the design of a capillary-compensated bearing may be used in an analogous manner.

## SUMMARY

This analysis shows the essential parameters involved for predicting and analyzing the performance of both orifice- and capillary-compensated hydrostatic journal bearings. The effect of accounting for circumferential flow is illustrated for both types of bearings, and is shown to be appreciable when the pocket aspect ratio is relatively large. Orifice-compensated bearings are shown to exhibit a load capacity superior to the capillary-compensated bearing. The effect of the type of loading is discussed. An approximate theoretical treatment of the influence of journal rotation is presented, and indicates that load capacities based on static conditions may be conservative for rotating bearings.

It should be emphasized that the analysis given in this paper for both the orifice- and capillary-compensated bearings applies only if the flow in the various bearing passageways is laminar. Since the use of low viscosity lubricants can result in turbulent flow through these passageways, an extension of the present analysis is necessary to accurately treat this case. The authors hope to make this the subject of a future paper.

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# Personals

Jefferson Chemical Co., Inc., N. Y., N. Y., has announced the promotions of A. K. May as Manager of Sales Research & Development, and J. R. Carlson & R. E. Werley as Eastern & Midwestern Regional Managers, respectively.

W. C. Willits, formerly with Kidder-Peabody & Co., has been appointed Assistant to the President of Pennsalt Chemicals, Phila., Pa.

Cuno Engineering Corp., Meriden, Conn., has announced the following appointments: **G. C. Whorf** (ASLE Connecticut Section), as Manager of their new Cincinnati, Ohio, branch office; J. E. McKay, as Regional Manager of Distributor Sales for their Connecticut Filter Div., and the J. R. Carlson Co. & E. C. George Co., as new sales representatives for the Indiana & Ohio areas, respectively.

K. G. Ellsworth, formerly with American Thread Co., has been appointed Director of Public Relations for the American Standards Assoc., N. Y., N. Y.

F. T. Goll has been promoted to Sales Manager of the C. A. Norgren Co., Englewood, Colo.

Chemicals & Materials Corp., Terre Haute, Ind., has announced the appointment of John H. VanMoss (Sr. & Jr.) Associates as their general sales representatives in the Chicago area with offices at 332 S. Michigan Ave., Chicago, Ill.

**J. R. Dingle** **S. L. Eastman** **L. O. Witzenburg**

The 1956 Richards Memorial Award was presented to E. M. Barber, Supervisor of Special Engine Research, Texaco Research Center, Beacon, N. Y., for "demonstrating outstanding achievement within 20 to 25 years following graduation from a regular mechanical engineering curriculum in a recognized college or university." Mr. Barber is the inventor of the Texaco Combustion Process, a new type of engine which operates without knock on fuel of any quality.

B. M. Holt, formerly with R. M. Parsons Co., has been appointed to the newly-created post of Project Director of the Planning & Development Dept. of American Potash & Chemical Corp., Los Angeles, Calif.

The Cleveland Worm & Gear Co. and its affiliate, The Farval Corp., Cleveland, Ohio, have announced the following appointments: **L. O. Witzenburg** (ASLE Cleveland Section), as Vice-President in Charge of Sales; J. R. Dingle, as Vice-President in Charge of Manufacturing; and S. L. Eastman, as Vice-President in Charge of Engineering.

# The Assessment of Automobile Bearing Performance From Field Tests

**The conditions to which bearings are subjected in service include wide variations in load and temperature, presence of abrasive particles, and complex cycles of starting, stopping, idling, and resting. These conditions are practically impossible to simulate in laboratory tests. The results of such tests cannot, therefore, be directly applied in practice, although they give valuable fundamental information.**

**On the other hand, data from general service experience are difficult to obtain and frequently are unreliable.**

**This gap can be filled by controlled service tests, in which test bearings are fitted in standard vehicle engines operating under normal conditions. The test bearings and crankshaft are subjected to careful measurement for observing wear, and to close examination for qualitative assessment of results.**

**By testing similar bearings in two different positions in the same engine and by repeating test designs in different engines, the statistical significance of observed differences can be assessed and reliable comparisons between materials can be obtained. Information on factors giving rise to high wear figures can also be obtained. Examples of test results are given.**

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**Glacier Metal Co., Ltd.**
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**Wembley, Middlesex, England**

## INTRODUCTION

### The Need for Reliable Tests

Progress in engineering and metallurgical fields is largely dependent on reliable assessment of the comparative performance of different designs and materials. The performance of bearings is a complex matter and is not dependent on any simple property. It might be summed up as the property of carrying the maximum load, without mechanical breakdown by fatigue or distortion, and without suffering more than the minimum amount of wear or corrosion of either mating service within the period required. Engine designers are continually seeking improved bearing life under more and more arduous conditions. Very many different materials and combinations of materials are manufactured and used for bearings, but in order to make a rational choice of these available materials and in order to provide a basis for research into new materials, it is essential to have a source of reliable performance data. However, no single satisfactory test for this complex property known as performance is available, and current methods of assessment are far from satisfactory. Nevertheless, properly designed field tests can be an extremely useful means of supplementing other data which are available, and this paper describes the scheme of testing used and some of the results obtained.

## POSSIBLE SOURCES OF DATA

### (a) Laboratory Tests

In one type of laboratory test devised to evaluate the specific properties of materials, no attempt is made to simulate an actual bearing; instead the material is tested in some simple form. Examples of this are the friction and wear tests,[1-2] and fatigue tests,[3-4] which were carried out on simple laboratory samples. In other types of laboratory tests, actual bearings are used and certain service conditions are imposed on the bearing. The examples of this type of test are those made by Ludicke,[5] by Roach,[6] and by Love, Forrester, and Burke.[7] Both types of laboratory tests are extremely useful in research of a fundamental character and in determining materials to be used for further tests, but tests under laboratory conditions cannot be relied upon as a basis for selecting materials for service conditions. Under ideal operating conditions, special bearing materials are hardly necessary at all,[5] and it is the capacity of a bearing material to tolerate departure from the ideal which determines its usefulness in practice. Many materials which give good results under laboratory conditions fail as practical bearing materials. For example, results of tests have been published which give good rating to high tin bronzes. This material, however, is quite unsuitable for main or big-end engine bearings unless a soft overlay plating is used. In practice, a bearing must tolerate large variations in load, frequent starting and stopping, wide temperature variations, the action of a wide variety of lubricants, and the presence of a considerable variety of abrasive particles. Such conditions are extremely difficult to simulate in laboratory tests.

### (b) Engine Tests

Engine tests sacrifice to some extent the control of conditions which is obtained in laboratory rig tests and do

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not fully satisfy the need for simulation of service conditions. While being an essential form of tests for engine builders, from the point of view of manufacturers of a single component they are an unsatisfactory compromise. For practical reasons they must generally be of short duration, and again it is difficult to reproduce starting and stopping.

(c) Service Experience

General service experience is extremely useful in guiding the bearing manufacturer or engine builder in choice of the bearing design and material, but the data available are generally not sufficiently reliable to constitute a satisfactory basis for research and development work. Furthermore, it is impracticable to assess the performance of completely new materials by putting them into general service, for the results of an experiment on a very bad material might be catastrophic. The difficulty of keeping adequate records of the performance of vehicles in service has been experienced in other fields.[8]

(d) Controlled Service Tests

In view of the limitations of these three sources of information, the Company with which the authors are associated has developed a scheme of testing bearing designs and materials on automobiles used by the company's staff. This scheme had to be carefully designed in order to avoid misleading results, and has certain limitations, but within these limitations the results obtained have provided great assistance in research and development work being carried out. There is nothing fundamentally new in using such tests for engineering components, but so far as the authors are aware it has not been previously used on a systematic basis for bearings.

The major limitation of such tests is that they can seldom, if ever, be carried to the point of complete failure unless, of course, the material tested is most unsuitable for bearings. The normal life expected of a bearing under average conditions of service is at least several years, and research based on such long term tests would necessarily be very slow. Accordingly, this type of test is used only for obtaining data on factors such as wear resistance, score resistance, and other factors which can be assessed on the basis of fairly short term tests. Fatigue data are obtained from rig tests which can be accelerated by imposing loads considerably above those expected in normal operation. Some of the apparatus used by the authors for this purpose has already been described.[7]

## SCHEME OF FIELD TESTING

Normally the cars chosen for testing are selected from those which undergo at least 10,000 miles service per year and, preferably, considerably more. As wide as possible a variety of engine types is chosen. In general, a particular lubricant is specified; but apart from this, no attempt is made to prescribe the way in which the car is to be used. The user of the car is merely asked to drive it normally for his ordinary purposes.

It has been found desirable to make a practice of using either new or reconditioned engines for test purposes. A considerable amount of detritus (loose particles) is practically always found in new or reconditioned engines, derived presumably from swarf (metal grit) which, despite normal precautions, always remains to some extent in the engine after assembly, and from the abnormal wear which occurs in other surfaces during the running-in stages. If new bearings are fitted to an engine which has run for some time, such bearings normally will suffer much less damage than is experienced with bearings fitted initially into an engine.

Since no attempt is made to simulate service conditions, it follows that tests are only comparative within themselves. It is, therefore, necessary to include two or more different types of bearing in an engine in order that the test shall have any real value. It is also usually desirable to have at least two specimens of a given design and material in the same engine in order that an estimate can be obtained of the variability of the test and, hence, of the significance of any comparative results which are obtained.

It is not an easy matter to decide the proper duration of this type of test. The arguments for short-term tests are that the major wear and scoring generally occurs during the first stages of engine life, and the shorter the time of test the more rapidly can information be gained. On the other hand, tests cannot be cut so short as to be misleading. Normal practice has been to standardize on 13,000 miles as the duration of the normal test but, in addition, all test bearings are examined after 3,000 miles. Also longer term tests are being carried out.

The normal test procedure adopted is as follows:

(a) Decide on the question to be answered, e.g. what is the relative wear performance of materials A, B, and C?

(b) Decide on a suitable vehicle for the test.

(c) Decide on a suitable design of test, e.g. if a four-cylinder engine is available, how the different specifications will be positioned in this engine and which specifications shall be duplicated in order to provide an estimate of variability.

(d) If the vehicle used has a new engine, fit the bearings before any running mileage is achieved; otherwise recondition the old engine under normal workshop conditions.

(e) Measure as accurately as possible all the relevant dimensions of the bearings to be used and the diameter and ovality of the newly ground crankshaft.

(f) Assemble the crankshaft and bearings into the engine and pass over to the normal user of the vehicle.

(g) After about 3,000 miles service, examine the bearing carefully and assess degree of surface damage visually.

(h) Re-assemble and run for further 10,000 miles.

(i) Strip down engine. Measure crankshaft and bearings and subject them to searching examination.

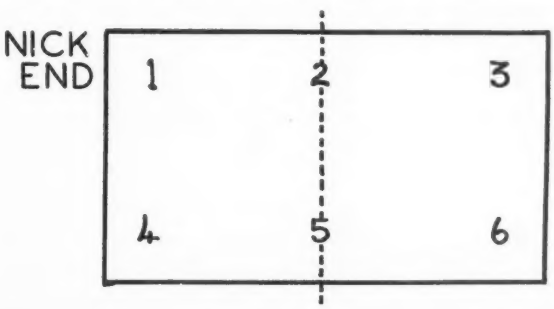


**Figure 1. Positions of test bearings as viewed from the back.**

Table I

Dimensions (in inches) of Crankpins Before and After Test

| | 1 | | | 2 | | | 3 | | | 4 | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Against Whitemetal | | | Against 55/45 Copper Lead | | | Against 70/30 Copper Lead | | | Against 55/45 Copper Lead | | |
| Position | Before | After | Difference | Before | After | Difference | Before | After | Difference | Before | After | Difference |
| 1 | 1.6250 | 1.6234 | .0016 | 1.6248 | 1.6240 | .0008 | 1.6246 | 1.6216 | .0030 | 1.6250 | 1.6241 | .0009 |
| 2 | 1.6249 | 1.6246 | .0003 | 1.6248 | 1.6218 | .0030 | 1.6249 | 1.6237 | .0012 | 1.6250 | 1.6222 | .0028 |
| 3 | 1.6249 | 1.6247 | .0002 | 1.6248 | 1.6243 | .0005 | 1.6247 | 1.6235 | .0012 | 1.6251 | 1.6248 | .0003 |
| 4 | 1.6249 | 1.6247 | .0002 | 1.6248 | 1.6238 | .0010 | 1.6249 | 1.6242 | .0007 | 1.6251 | 1.6239 | .0012 |
| Total | 6.4997 | 6.4974 | .0023 | 6.4992 | 6.4939 | .0053 | 6.4991 | 6.4930 | .0061 | 6.5002 | 6.4950 | .0052 |
| Mean Wear | | .00057 | | | .00132 | | | .00153 | | | .00130 | |

Table II

Wall Thickness (in inches) of Bearings Before and After Test

| Shaft Position No. | 1 | | | | 2 | | | | 3 | | | | 4 | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Specification | Whitemetal | | | | 55/45 Copper Lead | | | | 70/30 Copper Lead | | | | 55/45 Copper Lead | | | |
| Stage | Before | | After | | Before | | After | | Before | | After | | Before | | After | |
| Bearing No. | 645 | 646 | 645 | 646 | 647 | 648 | 647 | 648 | 649 | 650 | 649 | 650 | 651 | 652 | 651 | 652 |
| Dimensions in inches 1 | .0520 | .0519 | .0519 | .0520 | .0519 | .0520 | .0514 | .0512 | .0519 | .0520 | .0513 | .0518 | .0521 | .0521 | .0512 | .0515 |
| 2 | .0520 | .0520 | .0520 | .0520 | .0520 | .0520 | .0517 | .0516 | .0520 | .0520 | .0518 | .0518 | .0520 | .0521 | .0516 | .0517 |
| 3 | .0517 | .0519 | .0521 | .0520 | .0521 | .0520 | .0518 | .0516 | .0521 | .0521 | .0520 | .0517 | .0520 | .0518 | .0517 | .0515 |
| 4 | .0520 | .0520 | .0520 | .0520 | .0520 | .0520 | .0510 | .0516 | .0520 | ..... | .0514 | .0517 | .0520 | .0520 | .0511 | .0513 |
| 5 | .0520 | .0520 | .0520 | .0520 | .0520 | .0520 | .0515 | .0515 | .0520 | ..... | .0517 | .0518 | .0520 | .0520 | .0516 | .0515 |
| 6 | .0520 | .0521 | .0521 | .0521 | .0521 | .0521 | .0517 | .0514 | .0521 | ..... | .0518 | .0518 | .0520 | .0520 | .0517 | .0515 |
| Mean Wear | (Mean gain) | | .00005 | | | .00052 | | | | .00032 | | | | | .00052 | |

## MEASUREMENT OF DIMENSIONS

### (a) Bearings

Before starting the test, the test bearings are examined in the six positions indicated by Figure 1 by a vernier micrometer which is standardized against the slip gauge nearest in dimensions to the wall thickness of the bearing.

The same six positions are measured again at the 3,000-mile examination and, of course, at the completion of the test.

All measurements are made by the same observer using the same micrometer.

The mean wear experienced by the bearings is taken as the mean of the initial six readings compared with the mean of the final six readings. Statistical studies have shown that the standard deviation of a single measurement with a vernier micrometer is $0.3 \times 10^{-4}$ in., and thus the standard deviation of the mean wear, which is the difference between two sets of six readings, is

$$0.3 \sqrt{\left(\frac{1}{6}+\frac{1}{6}\right)} \times 10^{-4}, \quad \text{or} \quad 0.2 \times 10^{-4} \text{ in.}$$

Different observers differ significantly in the accuracy with which they can measure so that it is important to use the same observer. No change of observer error with time has been detected.

The change in bearing weight is not considered because this has been found to have no correlation with dimensional change.

### (b) Shaft

Each crankpin is measured in the following four places, also by a vernier micrometer:

Diameter in centerline plane, i.e. adjacent to oil hole. Front.

Diameter in centerline plane, i.e. adjacent to oil hole. Rear.

Diameter across the centerline plane, i.e. at the right angles to the oil hole. Front.

Diameter across the centerline plane, i.e. at the right angles to the oil hole. Rear.

The mean of the initial four readings is compared with the mean of the four readings taken on completion of the test. Again the same observer is used.

The standard deviation of a single measurement is $1.6 \times 10^{-4}$ in. and, thus, the standard deviation of the mean wear measurements is

$$1.6 \sqrt{\left(\frac{1}{4}+\frac{1}{4}\right)} \times 10^{-4}, \quad \text{or} \quad 1.1 \times 10^{-4} \text{ in.}$$

A typical dimensional statement on a test is shown in Tables I and II.

## VALIDITY OF METHOD

The mean bearing and shaft wear figures for the different specifications tested in a particular engine are compared between themselves in Table III, and the relative differences taken as indicative of the relative merits of the materials.

It thus appears that whitemetal is much the superior bearing material. It causes less shaft wear and suffers less bearing wear. It also appears that of the two copper lead specifications, the 55/45 is the better one in respect of shaft wear but the poorer in respect of bearing wear.

Table III

Mean Bearing and Shaft Wear Data
(Dimensions in .001 inches)

| Material | Whitemetal (tin base) | 55/45 Copper Lead | 70/30 Copper Lead | 55/45 Copper Lead |
|---|---|---|---|---|
| Shaft Wear | 0.57 | 1.32 | 1.53 | 1.30 |
| Bearing Wear | 0.05 (gain) | 0.52 | 0.32 | 0.52 |

Table IV

An Indication of the Significance of Crankpin and Bearing Wear Data for Any Given Test (Dimensions in .0001 inches)

| Test No. | Crankpin Wear | | Mean Bearing Wear | |
|---|---|---|---|---|
| | Position 1 | 2 | Position 1 | 2 |
| 33/3 | 14.6 | 15.3 | 4.1 | 3.6 |
| 59/1 | 6.0 | 6.0 | 0.5 | 0.4 |
| 62/1 | 5.7 | 5.3 | 6.0 | 7.0 |
| 69/1 | 7.0 | 8.2 | 0.8 | 0.6 |
| 73/1 | 0.3 | 0.5 | –0.5 | –0.5 |
| 74/1 | 0 | 0 | 0.6 | 0.9 |
| 91/1 | 7.0 | 7.0 | 1.0 | 0.9 |
| 110/1 | 3.0 | 2.3 | 1.4 | 2.7 |
| 158/1 | 3.7 | 2.8 | 0.9 | 1.0 |
| 182/1 | 1.8 | 2.0 | –0.2 | 0 |
| R.R.70 | 13.2 | 13.0 | 5.2 | 5.3 |

What is the validity of this statement? The bearings were tested in different positions. What are the chances that the observed differences in wear, which are significantly greater than the measuring error, are due to the position in the engine and not to the bearing material?

To answer this question many duplicate tests have been carried out with bearings of the same specification in different combinations of the four positions, and the results are given in Table IV. The dimensions are in tenths of thousandths of an inch.

Not more than two crankpins were fitted with bearings of the same specification because the other crankpins were needed for experimental specifications.

It will be noted that the results differ very greatly from one test to another, but the results for the two positions are, in any one test, very reproducible. Statistical tests show that the differences between the mean shaft wear and mean bearing wear figures given in Table III for white-metal, 55/45 copper lead, and 70/30 copper lead are, in fact, statistically significant and can be used as a basis for valid conclusions.

It should be noted that the above degree of reproducibility between the results obtained on different crankpins in the same engine is only true of the big-end positions. There is not the same reproducibility between the main bearing positions. Accordingly, the test results given below apply only to bearings tested in the big-end positions.

## THE INFLUENCE OF DIRT

It was stated earlier that it was necessary to carry out these tests with either a new or a rebored engine in order to ensure that the conditions as regards contamination by steel swarf were sufficiently severe to give a significant result.

The necessity for this condition is illustrated below. The crankshaft wear suffered in a new engine in which the original shaft and bearings were replaced prior to the test, but the engine, not rebored, was 0.6 tenths of a thou. Subsequently, the engine was rebored and bearings of the same specification fitted. The crankshaft wear then recorded was 4.5 tenths of a thou.

Further evidence of the influence of dirt contamination is obtained from the metallographic examination of the bearings after use. Many steel and some non-metallic particles are found embedded in the surface, and presence of steel can be shown most graphically by the use of an "iron printing" technique in which a filter paper moistened with potassium ferricyanide solution is pressed onto the bearing surface. The presence of steel is indicated by the blue spots which appear on the filter paper.

A number of typical iron prints are shown in Figure 2, and it will be noted that in some the steel particles are distributed uniformly while in others they are concentrated heavily to one end of the bearing. This phenomenon has been noted by Bidwell[9] and is attributed by him to a centrifuging of the particles in the oil. Another explanation is the occurrence of shaft deflection which limits the egress of particles from one end of the bearings.

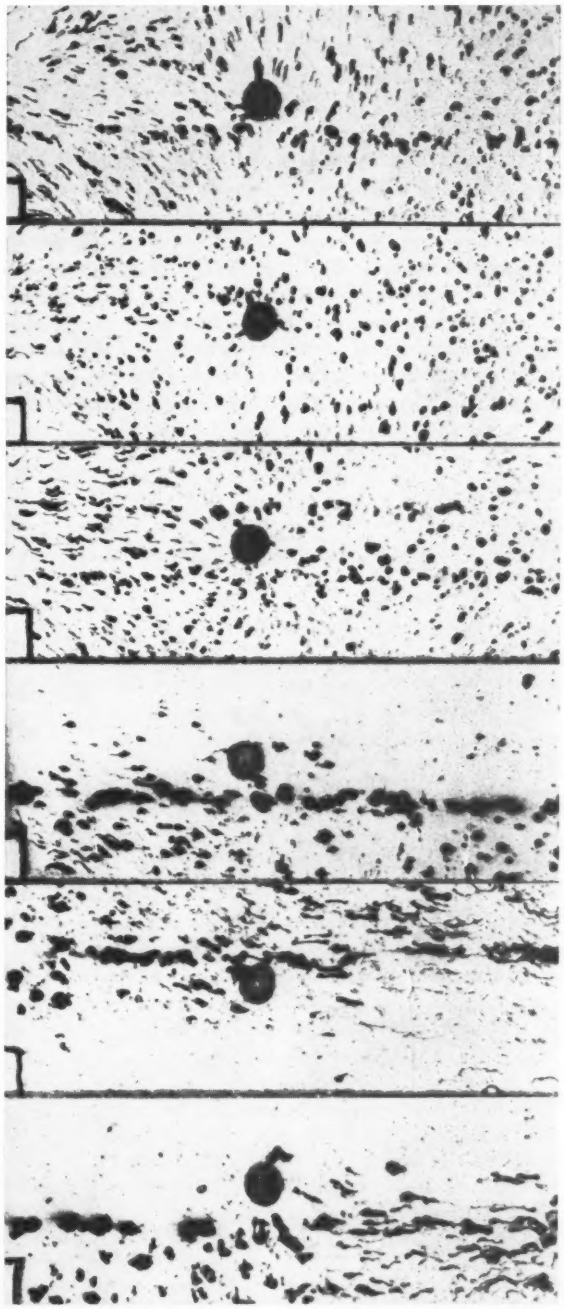

**Figure 2. Prints showing distribution of steel particles in the surface of used bearings.**

Table V

Effect of Particle Concentration on Shaft Wear
(Dimensions in .0001 inches)

| Shaft No. | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| Mean Wear* | 0.5 | 7.5 | 4.0 | 4.0 | 3.5 | 8.0 |
| Mean Wear† | 0 | 4.5 | 3.0 | 2.5 | 2.5 | 4.0 |

*End with the most particles.
†End with the least particles.

Table VI

Effect of Particle Concentration on Bearing Wall Thickness
(Dimensions in .0001 inches)

| Shaft No. | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| Mean Change* | +1.0 | +0.33 | +1.50 | +0.67 | +1.67 | +0.5 |
| Mean Change† | +0.33 | 0 | +0.80 | +0.17 | +0.17 | +0.33 |

*End with the most particles.
†End with the least particles.

Tables V and VI show shaft and bearing wear figures from a test in which this phenomenon of concentration of particles to one end of the bearings occurred. It will be noted that greater shaft wear had occurred at the end where the particles were concentrated, but the bearings had increased in wall thickness. Metallographic examination showed that this apparent increase in wall thickness was due to the presence of embedded particles, and this is one reason why there is no correlation between the dimensional change of a bearing and its change in weight.

## VARIABILITY BETWEEN VEHICLES

The results given in Table II for the differences between whitemetal and copper lead are striking. The 70/30 copper lead specification caused three times as much shaft wear as whitemetal. Differences of this order have not been obtained with every test, however. Several tests carried out with whitemetal and copper lead have shown very little difference between the two materials.

It has become apparent that some vehicles are "discriminating" in that they show quite large differences between different materials, and others are "non-discriminating." The results obtained from non-discriminating vehicles at one time threw doubt on the validity of the tests obtained from the discriminating vehicles, because it was difficult to know on which tests to rely.

For instance, one series of tests suggested that copper lead was much worse than whitemetal, and another that it was comparable with whitemetal. Which was to be believed? As stressed earlier, bearing materials are judged, not by the favorable circumstances in which they will operate, but by the unfavorable ones. It was clear that the discriminating vehicles were the ones in which the general level of wear was higher, due to greater dirt contamination, higher peak loads, greater stroke to bore ratios, more shaft deflections, and so on. Also, it was soon realized that, although the discrimination between materials differed from vehicle to vehicle, the direction of the difference was remarkably consistent.

This is illustrated in Table VII for a material which gives more shaft wear than whitemetal, i.e. copper lead, and for a material which gives less shaft wear than whitemetal, i.e. a thin overlay of lead/10% tin on copper lead.

Table VII

Shaft Wear Results for Various Metals, and Direction of Results Relative to Whitemetal. (Results expressed in 0.0001" loss in wall thickness.)

| | Copper Lead | | Lead Tin | |
|---|---|---|---|---|
| Whitemetal | Wear | Direction | Wear | Direction |
| 4.5 | 5.5 | + | | |
| 6.0 | 8.2 | + | | |
| 4.3 | | | 2.8 | — |
| 4.5 | 5.5 | + | | |
| .6 | | | | |
| .5 | | | | |
| 4.3 | 6.8 | + | | |
| 1.5 | 1.8 | + | | |
| 2.8 | 3.0 | + | 1.8 | — |
| 7.5 | 14.0 | + | | |
| 4.5 | 4.7 | + | | |
| 1.3 | 2.0 | + | 1.2 | — |
| 6.5 | | | | |
| 5.6 | 14.0 | + | | |
| .6 | | | .8 | + |
| 3.0 | | | 3.0 | — |

The consistency of the positive sign in one case and the negative sign in the other indicates that although the results from systematic vehicle tests may vary in magnitude, they rarely vary in direction and, thus, are an effective way of classifying bearing materials for their suitability in service.

## GENERAL CONCLUSIONS

The experience gained so far with controlled service tests shows that they are capable of yielding results on the wear properties of bearing materials which are statistically significant and directly applicable to practice. Taken in conjunction with fatigue tests, they enable the fields of usefulness of current bearing materials to be assessed and also form a most useful tool in the development of new materials. Furthermore, the tests have provided information on the effect of the test conditions on the magnitude of the wear experienced.

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**It's DETROIT in '57** for the **ASLE 12th Annual Meeting!**

# The Use of an Automatic Viscometer for the Study of Lubricant Flow

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**Many lubricants, such as greases, are non-Newtonian and time-dependent materials. The flow behavior of such materials is studied from flow curves which are measured over a wide range of rates of shear and under fully controlled flow conditions of shear rate, applied time, temperature, and pressure. In this paper, these flow curves are measured with a rotational concentric-cylinder viscometer which is so designed that the flow curves can be recorded directly and that the flow conditions of shear rate and time can be controlled automatically. Flow curves, obtained under different controlled flow conditions, are presented for a diester oil and two lubricant greases. An analysis is made from the flow curves of the effects of the flow properties and their variations with time, rate of shear and temperature on bearing lubrication, and on the transport of lubricants in pipe lines.**

NOMENCLATURE

$C$ rate of shear correlation factor $C = \left\{ \frac{1 - \left(\frac{R_b}{R_c}\right)^{2N}}{N\left[1 - \left(\frac{R_b}{R_c}\right)^{2}\right]} \right\} \left(\frac{R_b + R_c}{2R_b}\right)^{2(N-1)}$

$D$ pipe line diameter
$f$ yield value
$G$ rate of shear
$h$ height of annulus
$K$ proportionality between $G$ and $n$
$k$ instrument constant $k = \frac{k_1}{k_2}$
$k_1$ instrument constant $k_1 = \frac{(R_b + R_c)^2}{16\pi}\left(\frac{1}{R_b^2} - \frac{1}{R_c^2}\right)$
$k_2$ instrument constant $k_2 = \frac{\pi h}{2}(R_b + R_c)^2$
$k_3$ instrument constant $k_3 = \frac{\left(\frac{1}{R_t^2} - \frac{1}{R_c^2}\right)}{4\pi h \ln\left(\frac{R_b}{R_c}\right)}$
$N$ structure number
$n$ rotational speed
$P$ average pressure per unit area
$R_b$ radius of inner cylinder, bob
$R_c$ radius of outer cylinder, cup
$T$ torque
$U$ plastic viscosity
$v$ mean flow velocity in the pipe line
$Z$ viscosity
$\eta$ apparent viscosity
$\theta$ time interval to change speed from zero to preset maximum
$\mu$ Newtonian viscosity
$\tau$ shearing stress
$\phi$ coefficient of friction
$\rho$ density

Subscripts:

$a$ shear-dependent
$t$ time-dependent
$o$ intercept

A knowledge of the flow properties of lubricants is essential for an understanding of many lubrication problems. In fluid film bearing lubrication, the coefficient of friction in the bearing is very dependent upon the flow properties. In the transport of lubricants in manufacture, feeding of bearings, and industrial applications, the flow properties are undoubtedly most important in determining the requirements for the pipe line design.

However, the knowledge of flow properties alone will not solve any lubrication problem. An understanding of the flow process during lubrication is most essential. A certain relationship between lubrication and flow behavior has been established for fluid film or hydrodynamic lubrication in bearings.[1-2] This relationship applies for Newtonian fluids. Since many lubricants are non-Newtonian, an attempt is made in this paper to analyze this relationship so that it can be applied to non-Newtonian lubricants. Pipe line flow has been analyzed for Newtonian and non-Newtonian materials.[3-7] The analysis given in References 3, 4, and 5 will be used in this paper in regard to the lubricant flow of non-Newtonian materials in pipe lines.

The flow properties are obtained from viscometer measurements. Since many lubricants are non-Newtonian materials, their flow properties will vary with the flow condition. Therefore, in many cases an extended range of measurements is required before a valid interpretation of the flow behavior of a non-Newtonian lubricant is possible. In this paper, a viscometer[8] which permits automatic and direct measurements will be described briefly, and a short discussion will be given on how to obtain the flow properties from these measurements and how to use them in lubrication processes.

For example, a flow analysis will be attempted of a lubricant oil and two lubricant greases for fluid film bearing lubrication and pipe line flow. The analysis is made

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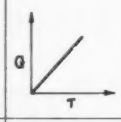
| MATERIAL | NEWTONIAN | BINGHAM PLASTIC | PSEUDO-PLASTIC | DILATANT | TIME-DEPENDENT |
|---|---|---|---|---|---|
| FLOW CURVES |  | | | | |
| FLOW CURVE EQUATIONS | $G = \frac{1}{\mu}\tau$ | $G = \frac{1}{U}(\tau - \tau_o)$ | $G \propto \tau^N$ | | |
| ROTATIONAL VISCOMETER $G = n/k_1C$ $\tau = T/k_2$ | $C = 1.00$ | $C = 1.00$ | $C > 1.00$ | $C < 1.00$ | $C \neq 1.00$ |
| FLOW PROPERTIES | $\mu$ | $U$ $f$ | $\eta_G$ $N > 1$ | $\eta_G$ $N < 1$ | $U_{G,t}$ $\eta_{G,t}$ or $f_{G,t}$ $N_{G,t}$ |
| ROTATIONAL VISCOMETER EQUATIONS | $\mu = k\frac{T}{n}$ | $U = k\frac{T-T_O}{n}$ $f = k_3T_O$ | $\eta_G = k\frac{T \cdot C}{n}$ $N = \frac{\log n}{\log T}$ | | |

**Figure 1. Flow curves and flow properties.**

from viscometer flow data obtained under different controlled flow and temperature conditions, and by considering the relationships for fluid film lubrication and pipe line flow.

## FLOW CURVES

The flow analysis of lubricants is best made from flow curves of rate of shear against shearing stress. This is suggested even for materials which are believed to be Newtonian, since some of those lubricants have shown unexpected anomalous behavior at extreme conditions of temperature, pressure, and shear rate. However, the flow of many lubricants depends on the applied rate of shear and, frequently, also on the time during which the shear rate is applied. Therefore, for a meaningful flow measurement of such materials, instantaneous indications of rate of shear and shearing stress over an extended range are required, and the flow curves have to be carefully programmed, preferably by automatic means, so that reproducible measurements can be obtained. For additional information, shearing stress-time curves are of value, since they show directly if there is a change in apparent viscosity with an increase in time during which a constant shear rate is applied. To be meaningful, all curves should be obtained at constant temperature, for laminar flow, and under conditions which permit control of the sequence and duration of the rate of shear.

The shape of the flow curve classifies the material in regard to its rheological behavior. In Figure 1, some of the most common flow curves are shown to demonstrate the difference in flow behavior of the various types of materials. Equations are assigned to some of these flow curves, from which flow properties can be obtained after the general equations are expressed in terms of the instrument with which the flow curve measurement is made. In Figure 1, only the equations for a concentric-cylinder rotational viscometer are presented, since this is the type of instrument used for the flow studies in this paper.

## AUTOMATIC VISCOMETER

The concentric-cylinder rotational viscometers[8-13] are well suited for the measurement of flow curves of non-Newtonian materials. The automatic concentric-cylinder rotational viscometer[14] built at the NACA Lewis Flight Propulsion Laboratory provides automatic means for increasing and decreasing the applied rates of shear, and also controls automatically the time that each shear rate is applied to the sample. This is important for highly time-dependent materials, since the flow properties frequently change so rapidly with time and shear rate that even the most careful manual control is inadequate in speed and sensitivity. Because of automatic controls, this viscometer can record a continuous flow curve in 15 seconds up to a rate of shear of about 4000 seconds$^{-1}$. Thus, the heat generated in the sample because of the shearing action is minimized.

The complete assembly of the automatic viscometer is shown in Figure 2, and the concentric-cylinders with their associated parts are shown in detail in Figure 3. In this viscometer, the outer cylinder, or cup, is rotated while the inner cylinder, or bob, is stationary, and the sample is sheared between the two. When the cup is rotated, the test sample exerts a torque, $T$, on the stationary bob which is measured almost instantaneously with a Statham strain gage transducer. As is seen in the block diagram (Figure 4), the cup is geared to a motor which drives the cup at speeds varying continuously from 0 to 1600 or

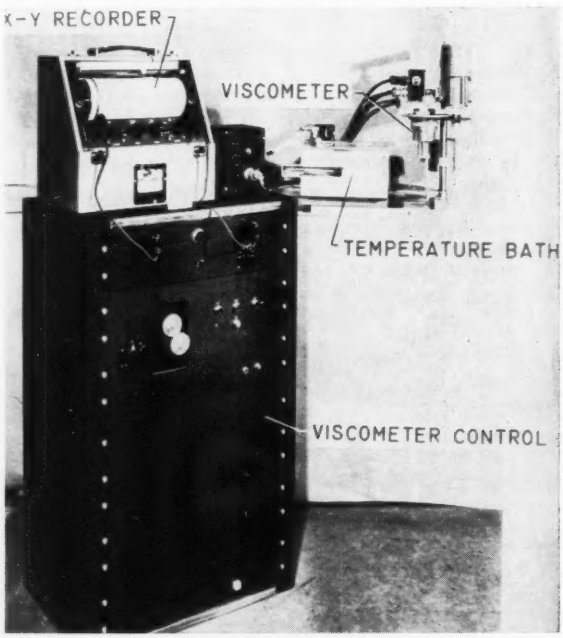



**Figure 2. Complete assembly of automatic viscometer.**

0 to 400 rpm, depending upon the selected gears. The motor is geared to a D-C tachometer and its speed, when subjected to varying loads, is feedback regulated. The cup and motor speeds are increased and decreased continuously by means of a motor driven potentiometer. Acceleration is constant within 1 percent and can be changed so that the time, $\theta$, required to increase the motor speed from zero to any selected maximum speed is 15, 30, 60, 120, or 240 seconds. The maximum speed is preset on a second potentiometer. The output of the tachometer is an instantaneous indication of the rotational speed, $n$, of the cup.

In a concentric-cylinder rotational viscometer, both the rate of shear, $G$, and the shearing stress, $\tau$, are inversely proportional to the square of the radius.[15] In this viscometer, with small separations of 0.03, 0.04 and 0.11 centimeter between cylinders, it is suggested[4] that the midpoint radius be used to calculate $G$ and $\tau$. Then $G = n/Ck_1$, and $\tau = T/k_2$ (Figure 1), where $C = 1.00$ for a Newtonian and a Bingham plastic, provided $\tau \geqq f$. For other non-Newtonian materials, $C$ depends on the design of the cylinders and on the test material.[4] The three cylinder sets of this viscometer have ratios $(R_b/R_c)$ of 0.90, 0.925, and 0.96. Thus, for most materials, it is possible to select a set of cylinders so that $C$ is between 0.95 and 1.05 for the sample to be tested. Therefore, $C$ can be assumed to be equal to 1 for most measurements. For Bingham plastic materials, complete flow across the separation between the cylinders is only obtained at a rate of shear at which $\tau \geqq f$. This rate of shear[14] is 0.042, 0.075, and 0.115 f/U second$^{-1}$ for the three cylinder sets employed with this viscometer. At rates of shear below these limits, the measurements are valid only after extensive corrections for plug flow[16] are applied. The flow properties and the rotational viscometer equations are given in Figure 1 for various types of materials. The equations for the instrument constants and for $C$ are indicated in the symbol list. All other dimensions of this viscometer can be found in Reference 14.

Other considerations in the design of this viscometer were to assure complete laminar flow, to eliminate end effects, to minimize temperature rise during shear, and to provide a wide-range viscometer with least power requirements. The range of the viscometer provides rates of shear up to 4100 seconds$^{-1}$, shearing stresses from 50 to 250,000 dynes per square centimeter, and permits viscosity measurements from 0.05 to 20,000 poises. Special features are provided to fill the viscometer without entrainment of air when testing grease-like materials and to keep the materials within the annular space up to the highest rotational speeds.

Flow curves are obtained directly by recording on an X-Y recorder the instantaneous torque values obtained from the strain gage transducer and the instantaneous values of rotational speed obtained from the tachometer. The recorded values are accurate within 1 percent of full scale, yielding viscosity measurements within 4 percent. This viscometer can be operated manually or automatically. For automatic operation, it is provided with two automatically controlled programs to give two types of flow curves. The two sequences of speed increase and decrease for the two programs A and B are shown at the left in Figure 5 as a function of time. The time interval $\theta$ is the time chosen to change the speed from zero to the preset maximum. The operator is given the choice at all encircled points of speed change either to wait or to continue immediately the sequence. This is done so that he can study the time-dependency of the material. The dashed line in program A indicates that this speed increase is not recorded. Program C, also shown at the left in Figure 5, is provided to give shearing stress-time curves. These

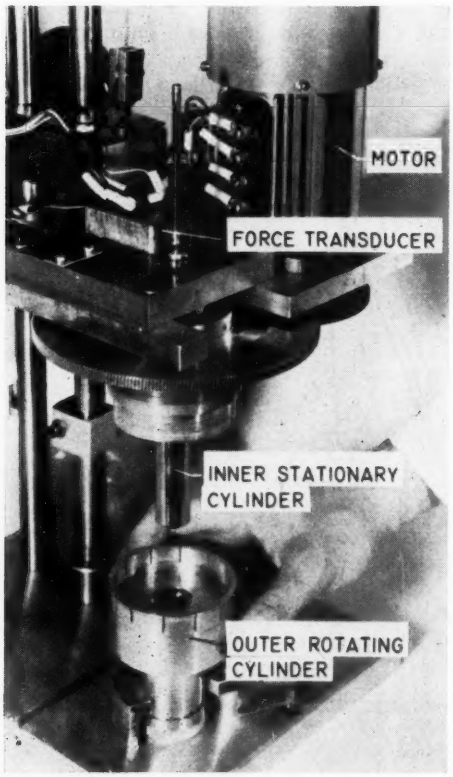



**Figure 3. Part assembly of automatic NACA viscometer.**

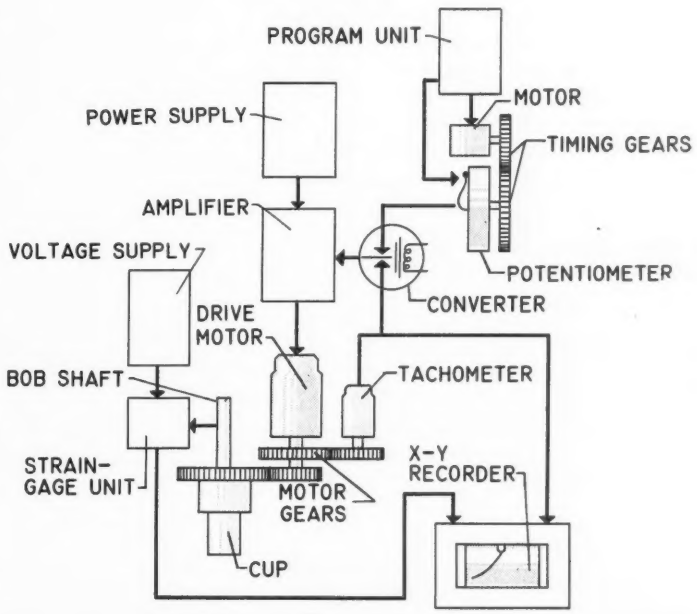



**Figure 4. Block diagram of automatic viscometer.**

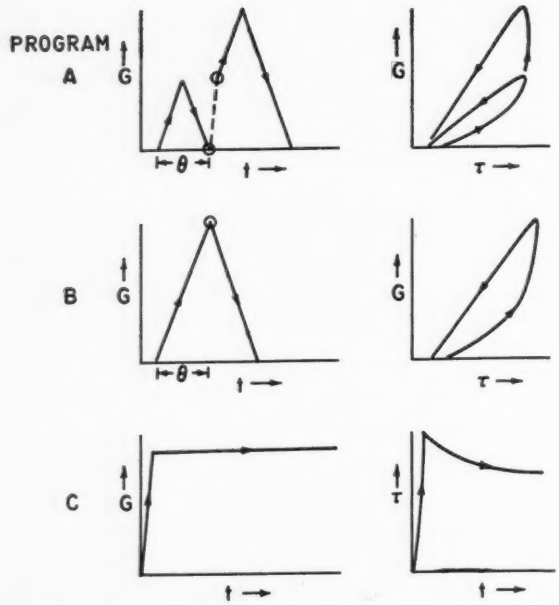



**Figure 5. Programs for the automatic viscometer and typical curves obtained with these programs for a time-dependent material.**

curves are obtained by recording a low frequency triangular wave instead of the tachometer output on the X-Y recorder. The sample curves to the right of the programs are typical for a time-dependent material. The flow curves follow the sequence of program A or B without interruption. The shearing stress-time curve is obtained at a constant rate of shear in accordance with the sequence *C*. The viscometer requires about one second to reach this rate of shear.

## FLOW BEHAVIOR

### Oil

A diester oil (MIL-L-7808) was measured on the automatic viscometer and its flow was found to be Newtonian at all temperatures from —10 to +40° C., up to rates of shear of 4000 seconds$^{-1}$. The viscosities of this oil are plotted against the reciprocal of the temperature in Figure 6.

### Greases

The flow behavior of two aircraft greases, AN-G15 (superseded by MIL-L-7711), and AN-G25 (superseded by MIL-G-3278), was studied. The first grease has a petroleum base, while the second one is compounded from a diester oil. Both greases show quite similar flow behavior. Unsheared samples of these greases, when measured with the automatic viscometer in accordance with program A, give flow curves similar to those shown in Figure 7. These particular flow curves were obtained for grease AN-G25 at a temperature of 0° C., and for a time, $\theta$, of 15 and 60 seconds. As seen from Figure 7, the flow structure decreased with an increase in time of shear. It was found from repeated measurements

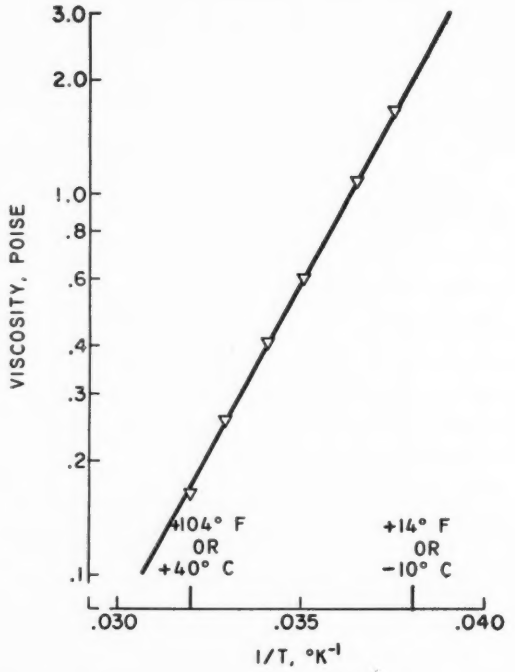



**Figure 6. Viscosity-temperature relationship for diester oil MIL-L-7808.**

that the flow structure did not recover even in a few months. The material was fully broken down at the maximum rate of shear of about 800 seconds$^{-1}$ after shearing it at this rate for about 300 seconds. This fully broken down curve did not change on additional shearing, unless the rate of shear or the temperature was increased. As is seen from Figure 8, different equilibrium curves were obtained for different maximum rates of shear, and the breakdown in structure increased with increasing maximum

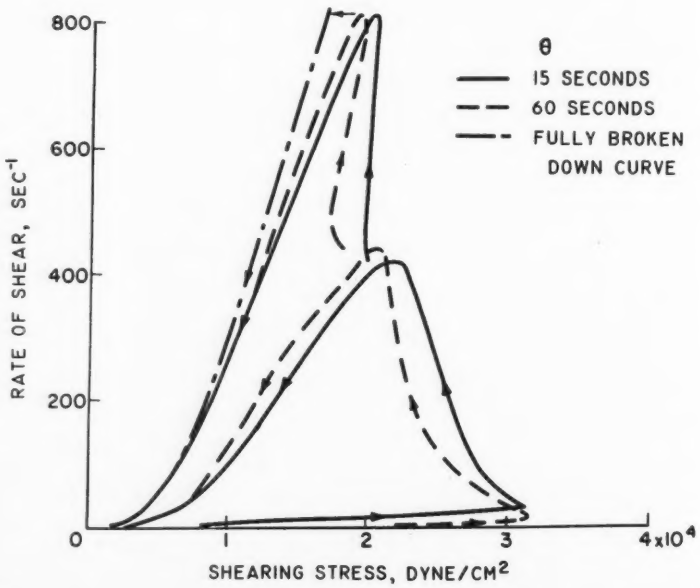



**Figure 7. Flow curves for grease AN-G25 using Program A. Temperature, 0° C.**

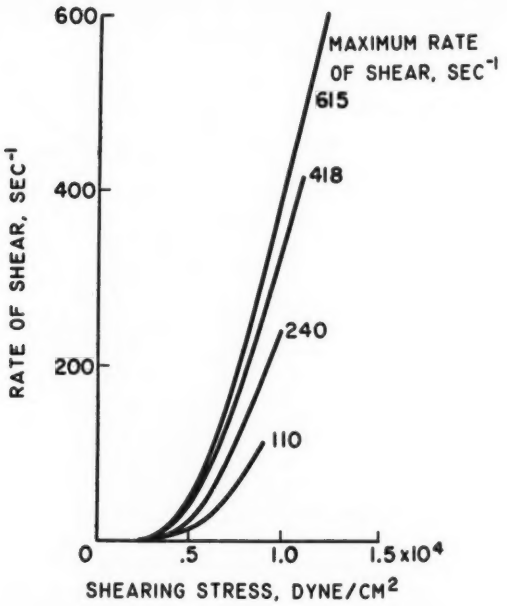



Figure 8. Down flow curves for grease AN-G25 after complete breakdown at different maximum rates of shear. Temperature, 0° C.; Program B; $\theta$, 30 seconds.

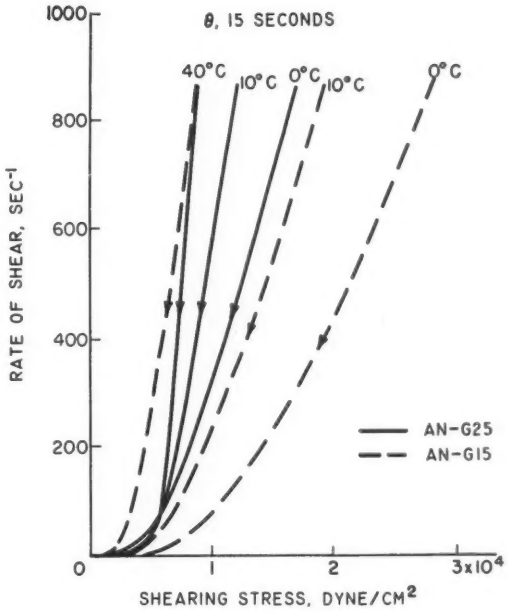



Figure 10. Down flow curves for greases AN-G25 and AN-G15 after complete breakdown at temperatures of 0, 10, and 40° C.

rates of shear. Figure 9 indicates the manner in which the shearing stress decreased when the material was sheared at a constant rate of shear. The equilibrium shearing stress seems to be reached in a somewhat shorter time at the higher rates of shear, but it never required more than 300 seconds of shearing to reach the equilibrium stress. Equilibrium curves obtained after shearing the sample at 850 seconds$^{-1}$ are shown in Figure 10 for the two greases and three temperatures each.

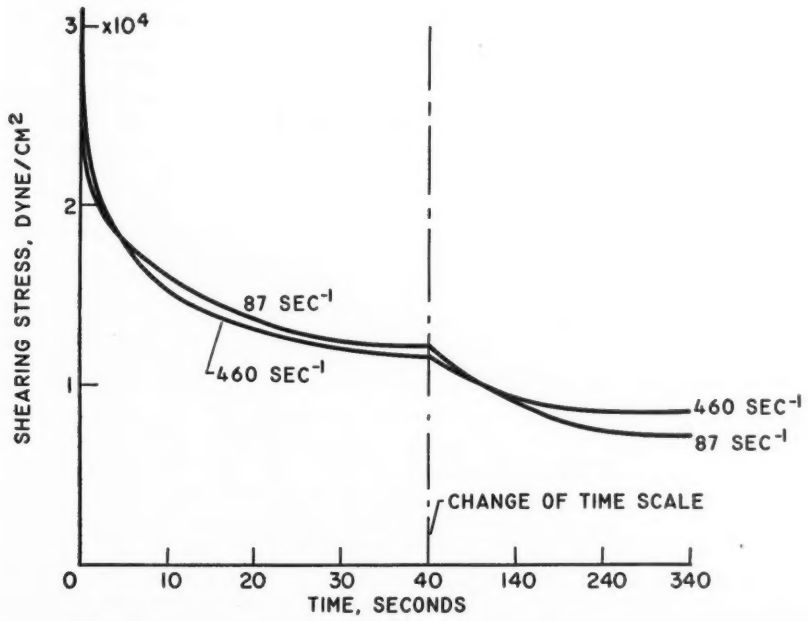



Figure 9. Shearing stress-time curves for grease AN-G25 at two constant rates of shear. Temperature, 25° C.; Program C.

Viscosities at temperatures other than those at which flow curves were measured, can be obtained from Figure 11 which gives the viscosity as a function of the reciprocal temperature but only for one rate of shear of 850 seconds$^{-1}$. For other rates of shear, additional flow-temperature data are required. Figure 11 indicates that, at this rate of shear, the apparent viscosity of grease AN-G15 is more temperature-sensitive than that of grease AN-G25. Since the equilibrium curves of grease AN-G25 (Figure 10) resemble curves for Bingham plastics, the plastic viscosity of this grease is also plotted in Figure 11 as a function of the reciprocal temperature. The yield value, $f$, seems constant in the temperature range from 0 to 40° C., and is about 6000 dynes per square centimeter.

For many applications, it is necessary to know if the grease will recover its structure on standing or by heating or cooling it. For this reason, repeated flow curve measurements were made on grease AN-G25 up to the same maximum rate of shear over a period of four months. It was found that the grease, once fully broken down in structure by the applied rate of shear, remained in this state of structural equilibrium when stored and measured at room temperature (about 25° C.). Temperature changes, however, had a marked effect on the flow structure (Figure 12). As is illustrated by experi-

ments A and B, some reformation of the structure of grease AN-G25 could be achieved by raising the temperature, and this reformation process seemed enhanced by shearing the grease at a raised temperature. When studying the flow curves obtained at 45° C. in experiment B, the build-up in structure is evidenced by the substantial yield value of the down-curve compared to that of the up-curve. Otherwise, the build-up in structure is detected by studying the flow curves measured at a lower temperature subsequent to the measurements at the raised temperature. Thus a breakdown in structure, which appears as a difference between the up- and down-flow curve measurements, indicates that build-up in structure must have occurred during the heating and cooling process. Experiment C indicates that no build-up in structure occurred when lowering the temperature.

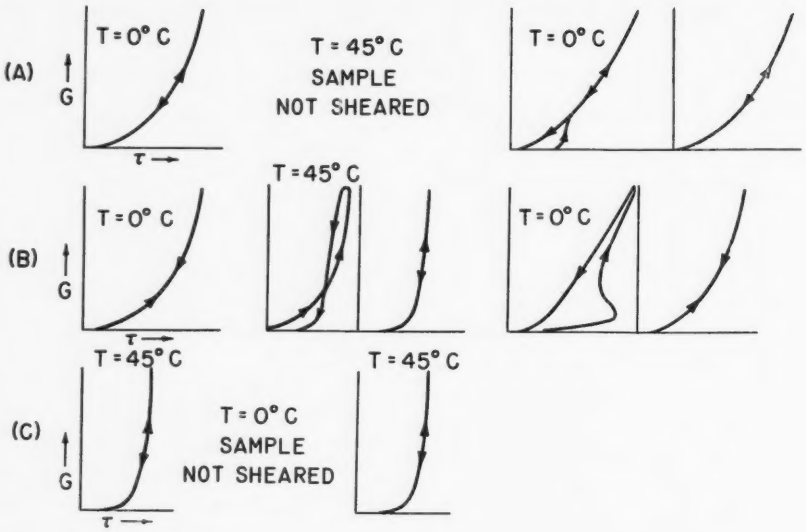



Figure 12. Change in flow structure of grease AN-G25.

The curves shown in Figures 7 to 12 are useful when a flow analysis is required for a specific purpose. However, to apply the measured flow properties to the study of a lubricant problem, it is essential to know the mechanics of flow during the lubrication process.

## BEARING FLOW ANALYSIS

In fluid film or hydrodynamic lubrication of bearings, the coefficient of friction depends greatly on the viscosity, while it is completely independent of viscosity in true boundary lubrication. This is indicated in Figure 13.[1-2] In this diagram, the coefficient of friction, $\phi$, is presented as a function of the bearing parameter $Zn/P$, where $Z$ is the viscosity, $n$ is the rotational speed, and $P$ is the average bearing pressure. Between the two regions, boundary friction on the left and fluid friction on the right, is a region of mixed friction where the fluid film is partially ruptured. In this region the coefficient of friction rises sharply with a decrease in bearing parameter.

Since $Z$ is the coefficient of viscosity, it is equal to the ratio of shearing stress, $\tau$, to rate of shear, $G$, and $G = Kn$. Then $\phi = \tau/P = K(Zn/P)$. Thus, the coefficient of friction is the friction force per unit area or shearing stress, $\tau$, divided by the average bearing pressure, $P$. In the region of fluid lubrication, the value of $K$ depends on the geometry of the sheared volume of the material, the flow pattern, and the separation of the shearing surfaces. If the viscosity can be considered constant across the separation, which is often the case because of the small separations usually encountered in lubrication applications, then $K$ will be constant for any one bearing, provided the bearing dimensions and the confines of the sheared sample remain constant.

Although fluid film friction factors for bearings can be calculated in some cases when the fluid film geometry permits the use of an established flow pattern, the points where fluid friction turns into mixed friction and mixed friction into boundary friction cannot be predicted by such calculations. These points are most frequently determined by experiment, which is set up to yield a friction curve similar to that shown in Figure 13.

The viscosity, $Z$, used in the bearing parameter is obtained from the flow curve; it is either the Newtonian or apparent viscosity. The Newtonian viscosity, $\mu$, is independent of shear rate and time, and depends only on temperature and pressure. However, the apparent viscosity, $\eta$ (Figure 1), is not only a function of temperature and pressure, but also of shear rate and frequently of time of shear. This means $\eta$

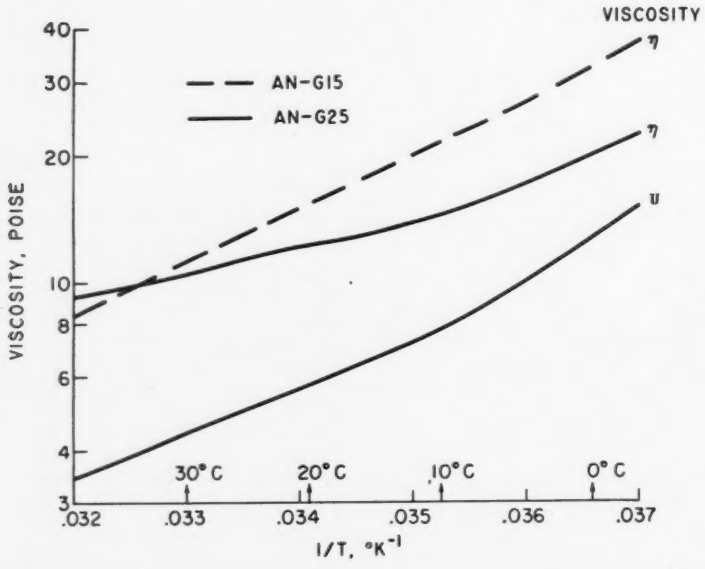



Figure 11. Viscosity-temperature relationship for greases AN-G25 and AN-G15, fully broken down at a maximum rate of shear of 850 sec$^{-1}$.

is a function of $K$ and, thus, can differ for the same $n$, if measured on two instruments which have different values of $K$. If the apparent viscosity is used in the bearing parameter, it is usually obtained from measurements with a viscometer that has a different $K$ than the lubrication tester which is used to obtain $\phi$. Therefore, $\eta$ has to be calculated from the viscometer measurements for the rate of shear $G = Kn$ which prevailed in the lubrication tester at the rotational speed at which $\phi$ was measured. In many cases when it is difficult to calculate the value of $K$ of the lubrication tester, an approximate shear rate can be estimated from the rotational speed and separation of the shearing surfaces, or $K$ can frequently be obtained from calibration with Newtonian fluids.

If the lubricant flow curve resembles that of a Bingham plastic, two flow properties which are the plastic viscosity, $U$, and the yield value, $f$, Figure 1, can be used instead of $\eta$. These two flow properties have the advantage that they are independent of shear rate and, therefore, correspond to any $n$ and $K$. When substituting them in the bearing parameter, $\eta = U + f/G$, and it is seen that $\eta$ approaches $U$ for high values of $G$ and can be very large for low values of $G$.

### Oil

The flow curves of the diester oil are straight lines indicating Newtonian behavior at all measured rates of shear up to 4000 seconds$^{-1}$. Since the shear rates in some lubricant applications are estimated[17] to be in order of $10^6$ seconds$^{-1}$, it might be interesting to consider the behavior of this oil at rates of shear of $10^6$ seconds$^{-1}$. To do that, a coefficient of friction curve similar to Figure 13 is required. In case that fluid flow exists at 4000 seconds$^{-1}$, the bearing parameter would be 250 times the value at $10^6$ seconds$^{-1}$ that it was at 4000 seconds$^{-1}$, assuming that laminar flow prevails and that the viscosity, temperature, and pressure remain constant. The friction factor at rates of shear of $10^6$ seconds$^{-1}$ is then obtained by extrapolating the coefficient of friction curve to the calculated bearing parameter, provided the separation does not change. Even if this oil becomes non-Newtonian at those rates of shear, its viscosity is unlikely to decrease by a factor of more than 250 which would be necessary before danger of mixed flow is present due to the viscosity decrease alone. However, a temperature rise in the fluid due to the high shear rate could decrease the viscosity sufficiently to cause mixed flow (Figure 6).

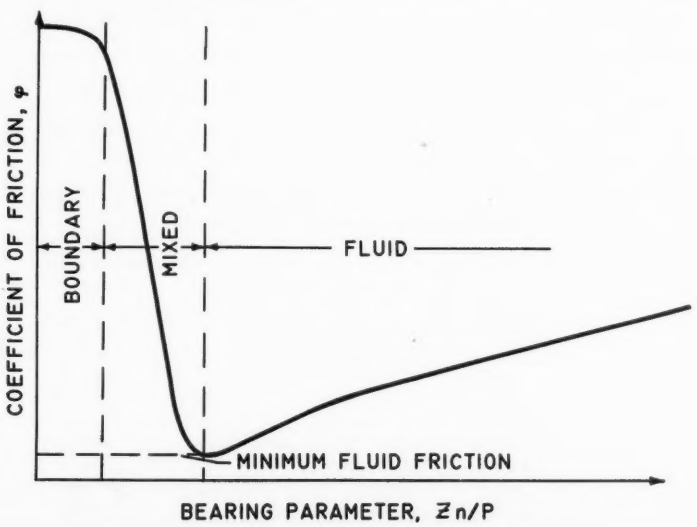



**Figure 13. Coefficient of friction curve for bearings.**

### Greases

If the non-Newtonian greases AN-G15 and AN-G25 are to be used as a lubricant in a bearing, a flow analysis would be required to set up a coefficient of friction curve to determine if fluid flow, mixed flow, or boundary lubrication will occur in the bearing under the given conditions of pressure, temperature, and rate of shear. To obtain the coefficient of friction curve, bearing parameters are required; and to calculate these for non-Newtonian lubricants, it is necessary to know the apparent viscosity at the same rates of shear, times of shear, temperatures, and pressures at which the coefficient of friction was measured. As previously indicated, the flow structure of those two greases breaks down when shear is applied and reaches an equilibrium structure in about 300 seconds. The structure depends on the maximum applied rate of shear (Figures 7 and 8), and does not change even when left unsheared for a few months. Therefore, it can be assumed that after the greases have been sheared for about 5 minutes, they will be broken down in the bearing or lubrication tester to an equilibrium structure which corresponds to the rate of shear in the bearing. Thus, frequently, unless starting up is considered, the viscosities can be taken from the respective equilibrium curves (Figure 8). The apparent viscosity as a function of rate of shear is then obtained from the end points of curves such as shown in Figure 8, or from the equilibrium stress of curves such as shown in Figure 9. Should the bearing be run in at a higher rate of shear and then be used continuously at a lower rate of shear, the apparent viscosity would have to be obtained from the equilibrium curve corresponding to the higher rate of shear but at the point of the lower rate of shear. In the case where starting up is considered and the grease is sheared less than 5 minutes at any one temperature, the shear history of the sample has to be taken into account as in Figures 7 and 12. Since $\eta = U + f/G$, the high yield value of 6000 dynes per square centimeter of grease AN-G25 indicates a relatively large $\eta$ and, thus, a large bearing parameter at low rates of shear. The use of flow properties $U$ and $f$ instead of $\eta$ has the advantage that they are shear independent along each equilibrium curve. For a complete analysis in regard to bearing lubrication, flow curves at various pressures would also be required, but they were not obtained in this study.

## PIPE LINE FLOW ANALYSIS

In transport problems through pipe lines, the friction factor and, therefore, the pressure loss depend greatly on the flow properties of the material. When dealing with pseudoplastic materials, the apparent viscosity and the structure number, $N$ (Figure 1), can be used to determine the

friction factor in the pipe line.[4-5] Since the apparent viscosity, $\eta$, is rate of shear dependent, it has to be calculated for the particular flow condition or rate of shear which exists in the pipe line. For a Bingham plastic, $U$ and $f$ are used to determine the friction factor.[3-5] Since they are independent of rate of shear, the flow condition in the pipe line does not have to be determined unless these materials are time-dependent, in which case the flow properties have to be calculated for the rate and time of shear in the pipe line.

### Oil

Since the diester oil is a Newtonian fluid, the pressure losses in pipe lines are a function of the viscosity alone. They are determined for laminar and turbulent flow by using the friction diagram for pipe lines[18] from which the friction factor can be obtained as a function of the Reynolds number, which is equal to $Dv\rho/\mu$.

### Greases

If greases AN-G15 and AN-G25 are to be transported through pipe lines, a flow analysis is required to determine the pressure loss in the pipe line from the friction factor. This friction factor can be obtained for any one flow condition, if the two flow properties of the grease are known at that same flow condition. From Figure 10, it seems that both greases show pseudoplastic behavior. Thus the two flow properties $\eta$ and $N$ can be used (Figure 1). If the rate of shear which exists in the pipe line at the flow rate for which the pressure loss is to be determined is the maximum rate of shear to which the grease was subjected, the apparent viscosity is obtained from the end point of the equilibrium curve (Figure 8), which corresponds to the rate of shear in the pipe line and which is $G_p = 2v(N+3)/D$ in laminar flow. Should the greases have been subjected to a higher rate of shear before transport than that corresponding to the flow rate in the pipe line, the apparent viscosity would have to be obtained from the equilibrium curve corresponding to the higher rate of shear but at the point of the rate of shear in the pipe line, $G_p$. Under other flow conditions, the shear history (Figures 7 and 12) might have to be considered to calculate an appropriate apparent viscosity. This apparent viscosity is then used to calculate a Reynolds number which is equal to $Dv\rho/\eta$, and the friction factor can be obtained as a function of this Reynolds number and the structure number, $N$, from a friction diagram.[4-5] Both flow properties, of course, are obtained for the temperature which the material is expected to have in the pipe line. The structure number, $N$, for grease AN-G15 seems to be constant in the temperature range from 0 to 40° C., and is equal to about 2.0 for the equilibrium curves obtained with a maximum rate of shear of 850 seconds$^{-1}$ (Figure 10), while the structure number, $N$, for grease AN-G25 for the equivalent flow curves, seems to increase almost linearly with temperature and ranges from 2.0 to 3.5 for the temperature range of 0 to 40° C.

The flow curves of grease AN-G25 resemble curves for Bingham plastics and, thus, can be analyzed as such. For Bingham plastics, the friction factor in pipe flow is a function of a Reynolds number and the plasticity number and can be obtained from a friction diagram.[3-5] The plastic viscosity of that equilibrium curve (Figure 8) corresponding to the maximum rate of shear to which the grease was subjected and to the temperature in the pipe line, is used in the Reynolds number which is equal to $Dv\rho/U$, and the plasticity number is calculated from flow properties, being equal to $fD/Uv$. The use of the Bingham plastic flow properties has the advantage that only the maximum shear rate to which the grease has been subjected either in the pipe line system or prior to its transport has to be known.

## SUMMARY

The paper analyzed briefly the lubrication problems with non-Newtonian materials in bearings and pipe lines. It also discussed the flow behavior and the measurement of flow properties of some non-Newtonian materials. From the discussion it becomes evident that flow curve measurements under fully controlled flow conditions are essential to give results when dealing with time-dependent materials, if the results are to be used to analyze lubrication problems. Such a flow analysis was attempted by using programmed flow curves obtained with an automatic viscometer. The flow and lubrication analysis is not complete and is only intended to lead the way to further studies of these problems. In fact, experimental data for bearing lubrication with non-Newtonian materials are seriously lacking. Much work remains to be done to assess fully the importance of the flow behavior of the lubricants for determining their performance in lubrication and transport.

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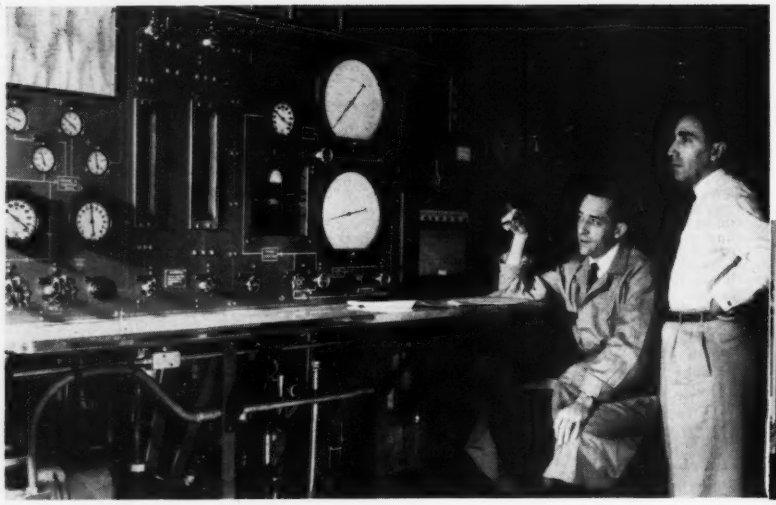
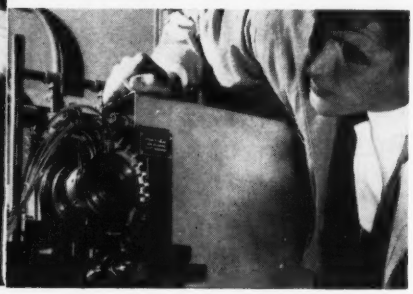


Journal bearing tests are monitored on this panel board containing more than a dozen instrument gauges and meters. R. E. Elwell, left, and Harry Apkarian, are studying the performance characteristics of large hydrodynamic journal bearings similar to those used in large steam turbines.



Engineer G. Fox runs a high-temperature test on a new hydrodynamic air bearing in General Electric's new Bearing and Lubricant Center. Air-lubricated bearings have special application, where extremely high temperatures are involved, in radioactive areas, or where extremely low friction is required.

## GE Bearing and Lubricant Research

By R. O. Fehr, Manager, Mechanical Engineering, General Engineering Laboratories.

"The new million-dollar GE Bearing and Lubricant Center is the most advanced engineering center of its type in the world. It represents the most intensive technological effort in the field.

"The problem of bearings and lubrication has been with people since the invention of the wheel. Those people used animal grease to reduce the friction between the wheel and the axle. When the wheel began to spin faster, the first major 'breakthrough' had to be made. We now find ourselves on the threshold of the age of space satellites, when another 'breakthrough' must come.

"We at General Electric are intensely interested in this technology because most of our products involve rotating equipment.

"One phase of our program for the immediate future is designed to increase the temperature range under which our equipment can operate. We feel that air bearings will make the 'breakthrough'. In addition, we are studying the possibilities of using molten metals and other materials for lubrication, which will enable us to go to higher speeds, higher temperatures, and work within radioactive atmospheres.

"Since the beginning of time the Earth has rotated without bearings and without lubrication, held in space by mass forces. Engineers are looking for ways to apply these principles to rotating equipment in the distant future. We are confident that these concepts will lead to dramatic technological changes, which may involve nonlubricated magnetic bearings held in place by magnetic forces. We are also confident that the facilities, engineering skill and know how of this new Bearing and Lubricant Center will make valuable and significant contributions toward these objectives."

Dr. Robert O. Fehr, left, manager of mechanical engineering for General Electric's General Engineering Laboratory, views a ball-bearing test project with Dr. Beno Sternlicht, technical director of the new million-dollar Bearing and Lubricant Center.

**SPECIAL FEATURE**

# L.E. Tours New GE Bearing

Dr. R. O. Fehr, Manager of Mechanical Engineering; Mr. Harry Apkarian (Chairman, Hudson-Mohawk Section), Manager of the new Center; and Dr. Beno Sternlicht (Hudson-Mohawk Section), Technical Director of the new Center were questioned about the activities to be pursued by the staff of the Center in their future and current work.

**Q. With what areas of investigation will GE's work be directed?**

A. We are working on lubricants today for jet engines with temperatures of 400° F maximum at the bearings, and we are currently testing bearing materials for use at 1500° F. We will be looking at bearing materials that will show better high-temperature characteristics.

**Q. What types of lubricants will be evaluated for this work?**

A. Synthetic diester types, silicones and hydrocarbon materials in various combinations of these. Silanes also are being investigated as high-temperature lubricants.

**Q. There has been considerable interest in the proper lubrication of low load, high speed applications under extreme environmental temperatures. Will GE's new facility deal with such applications?**

A. Yes.

**Q. What type of bearings can be anticipated for these applications?**

A. Ball bearings presently. Work is also being done on air-lubricated journal and thrust bearings.

**Q. What part will air, water, and liquid metals play in the development of lubricants for nuclear power applications?**

A. All will play an important part.

**Q. Do you visualize the use of other materials for this function?**

A. Yes. Effort is being made to use modifications of conventional lubricants, especially in the area of low radiation intensity.

**Q. GE has done some interesting things with special materials such as synthetic sapphires. Will these studies be extended to other minerals, and if so, with what objective in mind?**

**Functional Summary**

The new center is a part of the GE engineering laboratories, and combines work formerly carried out by the Thomson Laboratory at Lynn, Mass. and the General Engineering Laboratory at the Campbell Avenue Plant in Schenectady, N. Y.

Approximately 10,000 sq. ft. of floor space has been utilized in providing the area and equipment necessary for the 30 engineers, chemists, physicists, metallurgists, laboratory and engineering assistants which form the current staff.

Functions of lubricants as cooling media, ability to decrease friction and carry loads in either hydrostatic or hydrodynamic conditions; fundamental designs, including film thicknesses required, oil requirements, temperature rise, maximum temperatures encountered and materials required, static conditions and surface phenomena, stability conditions (vibration), turbulence, and fatigue are some of the areas of fundamental interest to be pursued.

The laboratory will serve all departments of GE, its divisions and groups, and will act as a center for lubrication and bearing information with unlimited scope not to be confined to conventional or even presently known materials or concepts. Laboratory functions will include training of lubrication engineers, collection of data to support studies on fluid films, frictional properties, and bearing materials. Bearings will be considered as any two surfaces moving relative to each other. The staff will draw upon consultants from private firms, educational institutions, and foreign countries when necessary.

Temperatures will range from —65° F to above 1500° F.

In addition to air, Helium, Nitrogen, and Argon will be considered as lubricants for study purposes.

Laboratory technician J. Foley momentarily invades a cloud of hot fumes to regulate oil flow during a high-temperature jet-engine bearings test. Ball bearings are tested under simulated jet-engine conditions at temperatures in excess of 500° F.

# and Lubricant Center

**by J. P. Thomas, Shell Oil Company, Albany, New York**

A. We are investigating many materials to uncover those most suitable for particular bearing application.

**Q. What role will the lubrication engineer play in the design of the new production equipment in the future? Have you observed any tendency toward greater acceptance by other engineering design groups to include the lubrication engineer as an integral part of the overall systems design before production design work commences?**

A. Greater acceptance of the lubrication engineering function has been recognized by integration of his services into the earlier design stages. This is one of the major factors recognized in setting up this laboratory. It is the most modern tool of the research lubrication engineer for investigating high temperatures and stresses at speeds above the supersonic ranges.

Particularly impressive to your reporter were the instrumentation and recording equipment. No data need be taken by the laboratory staff, and therefore, no possibility for error exists when reading gages, thermometers, and related instruments. Recording pressure and temperature gages were apparent as was a tremendous array of photographic equipment. I was impressed with the amount and quality of equipment and personnel assembled to study what is to be a very important sphere of engineering. Noticeable was the emphasis being placed on high-temperature work and the units for investigating fundamentals of lubrication and transient conditions of high-impact loading.

The new laboratory facility seems completely equipped to evaluate bearings, lubricants, and related mechanical functions. During the discussion it was stated that General Electric might or might not manufacture such lubricants as may be developed from work being done by the laboratory.

The ball bearings shown in the test are spinning at a rate faster than the speed of sound.

An experimental high-temperature lubricant is subjected to 800° F. A stop-watch and a "sight glass" to the piping are used to count the number of drops per minute to obtain flow measurement data. The small oven is heated with calrods packed in the insulation.

# ASLE 12th

# ANNUAL MEETING PROGRAM

**(Tentative)**

**April 15, 16, 17, 1957**
**Sheraton Cadillac Hotel**
**Detroit, Michigan**

## Monday, April 15, 1957

**9:00 A.M. Business Meeting**

**10:00-11:30 A.M.**

**SESSION 1A Wear of High Temperature Seals Material**

R. W. Blair, Chicago Rawhide Mfg. Co., Chicago, Ill., Organizing and Session Chairman

1. "Wear of Carbon Type Seal Material With Varied Graphite Content."
   Max A. Swikert, R. L. Johnson, Lewis Flight Propulsion Laboratories, Cleveland, Ohio
2. "Frictional Properties of High Temperature Seal Material."
   Joseph F. Cerness, Carol Pilarczyk, Douglas H. Strong, Cleveland Graphite Bronze Co., Cleveland, Ohio

**SESSION 1B Environmental Factors VS Lubrication Effects**

A. W. Lindert, Standard Oil (Indiana), Whiting, Ind., Organizing and Session Chairman

1. "Some Effects of Gamma Radiation on Commercial Lubricants."
   J. G. Carroll, S. R. Calish, California Research Co., Richmond, Calif.
2. "Effects of Environment on Rust Inhibitors and Oxidation Stability of Lubricating Oils."
   A. Mertwy, M. Trachtman, J. Messina, H. Gissor, Frankfort Arsenal

**SESSION 1C Automatic Lubrication Equipment**

E. M. Higgins, Master Lubricants Co., Philadelphia, Pa., Organizing Chairman, Donald H. Wilson, Continental Steel Co., Kokomo, Ind., Session Chairman

1. "Circulating Oil Systems."
   N. S. Robson, Trabon Engineering Corp., Cleveland Ohio
2. "Oil Mist Lubrication."
   D. G. Faust, C. A. Norgren Co., Englewood, Colorado

**SESSION 1D Lubrication Engineering Course**

**SESSION 1E Exhibits**

**2:00-4:30 P.M.**

**SESSION 2A Metal Working Fluids**

H. L. Shaw, Battelle Memorial Institute, Organizing and Session Chairman

1. "Stamping and Deep Drawing Lubricants — What are They?"
   Leon Salz, Magnus Chemical Co., Garwood, New Jersey
2. "Conversion Coatings Aid the Lubrication of Titanium."
   Earl White, P. O. Miller, Battelle Memorial Institute, Columbus, Ohio
3. "Test for Evaluating Cutting Fluids."
   H. W. Husa, W. L. Buckley, Standard Oil Co. of Indiana
4. "Laboratory Evaluation of Cutting Fluids."
   H. A. Hartung, Atlantic Refining Co., Philadelphia, Pa.

**SESSION 2B Hydrodynamic Bearings**

Fred Macks, Air Guide Engineering Laboratories, Vermilion, Ohio, Organizing and Session Chairman

1. "A Solution for the Finite Journal Bearing and its Application to Analysis and Design —III."
   A. A. Raimondi, John Boyd, Westinghouse Research Laboratories, Pittsburgh, Pa.
2. "The Effect of Bearing Deformation in Slider Bearing Lubrication."
   F. Osterle, E. Saibel, Carnegie Tech., Pittsburgh, Pa.
3. "Some Mechanical Considerations in the Design of Large Thrust Bearings."
   R. A. Baudry, G. E. Peterson, Westinghouse Electric, Pittsburgh, Pa.
4. "Solution of Reynolds and Energy Equations for Sector Thrust Bearings."
   B. Sternlicht, General Electric Co., Schenectady, New York

**SESSION 2C New Greases for Problem Application**

Harry W. McCulloch, Jr., Shell Oil Co., Wood River, Ill., Organizing Chairman, J. W. Lane, President, NLGI, Socony-Mobil Oil Co., Session Chairman

1. "Performance of an Inorganic Microgel Grease in Industrial Applications."
   J. A. Bell, Shell Oil Co.. New York, New York
2. "Service Experience with a New Multi-Purpose Industrial Grease."
   S. R. Calish, Jr., California Research Corp., Richmond, Calif.
3. "A New Thickener for Multi-Purpose Industrial Grease."
   W. L. Hayne, D. R. Oberlink, R. J. Rosscup, Standard Oil Co., (Indiana), Chicago, Ill.

**SESSION 2D Lubrication Engineering Course**

**SESSION 2E Exhibits**

## Tuesday, April 16, 1957

### 9:30-11:30 A.M.

**SESSION 3A Specifying Lubrication for Industry**

H. McCulloch, Jr., Shell Oil Co., Wood River, Ill., Organizing Chairman, A. R. Black, Shell Oil Co., New York, New York, Session Chairman

1. "LUBRICATION STANDARDS FOR INDUSTRIAL EQUIPMENT."
   S. O. Kimball, A. C. Spark Plug Co., Flint, Michigan
2. "LUBRICATION IN A GLASS CONTAINER MANUFACTURING PLANT."
   D. E. Grube, Owens-Illinois Glass Co., Toledo, Ohio
3. "LUBRICATION IN A CHEMICAL PLANT."
   R. W. Clark, E. I. du Pont de Nemours & Co., Wilmington, Delaware

**SESSION 3B Performance, Design and Operation of Rolling Contact Bearings**

A. A. Raimondi, Westinghouse Research Laboratories, Organizing and Session Chairman

1. "DESIGN AND OPERATION OF BALL BEARINGS FOR USE IN PRESSURIZED WATER SYSTEMS."
   P. R. Eklund, Westinghouse Electric Co., Pittsburgh, Pa.
2. "THE TORQUE AND WEAR OF WATER LUBRICATED BALL BEARINGS AS RELATED TO CERTAIN DESIGN PARAMETERS."
   W. H. Goldwaite, C. M. Allen, and D. W. Knoll, Battelle Institute, Columbus, Ohio
3. "ROLLING CONTACT BEARINGS FOR NON-LUBRICATING SERVICE."
   B. E. Murphy, J. K. Elbaum, Haynes-Stellite Co., Kokomo, Ind.

**SESSION 3C Lubrication Fundamentals**

Dr. E. G. Jackson, General Electric Co., Lynn, Mass., Organizing and Session Chairman

1. "THE NON-NEWTONIAN CHARACTERISTICS OF LUBRICATING OILS."
   Theodore W. Selby, General Motors Research, Warren, Michigan
2. "MEASUREMENTS ON POLYMER MODIFIED LUBRICATING OILS AT HIGH RATES OF SHEAR."
   Dr. Waldmir Philipoff, Franklin Institute
3. "EXPERIMENTAL INVESTIGATION OF THE CORROSIVE COMPONENT OF MECHANICAL WEAR."
   Raymond H. Baskey, F. C. Wagner and J. T. Burwell, Horizons, Inc., Cleveland, Ohio
4. "IS BOUNDARY LUBRICATION A MYTH?"
   W. Lewicki, Imperial College of Science and Tech nology, Dept. Chemical Engineering, London, England

**SESSION 3D Lubrication Engineering Course**

**SESSION 3E Exhibits**

### 2:00-4:30 P.M.

**SESSION 4A The Nature of Chemisorbed Lubricant Films**

E. M. Kipp, Session Chairman, Dr. Herman Reis, Standard Oil Co., of Indiana, Organizing Chairman

1. "CHEMICAL NATURE OF EXTREME PRESSURE LUBRICATION."
   Dr. G. Hugel, Institut Francais Du Petrole, Paris, France
2. "ADSORPTION AND FRICTION STUDIES OF LUBRICANT ADDITIVES."
   Arnold Miller, Borg-Warner Corp., Des Plaines, Ill.
3. "A THEORETICAL CRITERIA FOR THE EFFECTIVENESS OF A LUBRICANT FILM."
   Ernest Rabinowicz, M.I.T., Cambridge, Mass.
4. "RECENT RUSSIAN RESEARCH ON CHEMISORBED LUBRICANT ADDITIVES."
   Douglas Godfrey, California Research Corp., Richmond, Calif.
5. "APPROACH TO FRICTION AND BOUNDARY LUBRICATION FROM THE DISLOCATION THEORY OF FRICTION."
   John H. Dismant, Consultant Research Engineer, Grand Junction, Colorado

**SESSION 4B Automation Lubrication Practices in the Automotive Industry**

Roger Hendrick, Reynolds Metals Co., Richmond, Virginia, Organizing and Session Chairman

1. "STANDARDIZATION OF PLANT LUBRICANTS AND LUBRICATION PRACTICES."
   R. O. Kageff, Ford Motor Co., Detroit, Michigan
2. "INCREASED MACHINE PRODUCTIVITY DEMANDS WELL-DESIGNED LUBRICATION PRACTICES."
   Robert H. Guy, Detroit Transmission Division, General Motors Co., Ypsilanti, Michigan
3. "ESTABLISHING LUBRICATION PRACTICES IN AN AUTOMATED ENGINE PLANT."
   D. F. Miller, Engineering Division, Chrysler Corp., Detroit, Mich.

**SESSION 4C Rolling Contact Bearings**

J. H. Gustafson, Marlin Rockwell Corp., Organizing and Session Chairman

1. "GREASE LUBRICATION IN ULTRA HIGH LOAD ROLLER-CONTACT BEARINGS."
   J. B. Ancinelli, Shell Development Co., Emeryville, Calif.
2. "THE EFFECT OF L. RIBBON AND SPRING TYPE SEPARATORS ON THE STARTING TORQUE OF AN R4D BEARING."
   John C. Lawrence, Air Research Mfg. Co., Los Angeles, Calif.

**SESSION 4D Lubrication Engineering Course**

**SESSION 4E Exhibits**

**6:30-9:30 P.M. Cocktail Party and Annual Awards Banquet**

## Wednesday, April 17, 1957

### 9:30-11:30 A.M.

**SESSION 5A Requirement for Automotive Hypoid Gear Lubrication**

R. O. Kageff, Ford Motor Co., Detroit, Mich., Organizing and Session Chairman

1. "NEW DEVELOPMENTS IN GEAR LUBRICANTS."
   C. M. Heinen, Chrysler Corp., Detroit, Michigan
2. "PROBLEMS ASSOCIATED WITH MARKETING GEAR LUBRICANTS."
   A. A. Manteuffel and J. B. Stucker, Pure Oil Co., Crystal Lake, Ill.

3. "CRC Evaluation of New Gear Lubricants."
T. P. Sands, Monsanto Chemical Co., St. Louis, Mo.
4. "Field Problems in Hypoid Rear Axles."
J. E. Cardillo, Ford Motor Co., Dearborn, Mich.
5. "Chemical Additives for Hypoid Gears."
Dr. R. A. Asseff, Lubrizol Co., Cleveland, Ohio

**SESSION 5B Railroad Journal Bearings**

L. E. Hoyer, American Brake Shoe Co., Detroit, Mich., Organizing and Session Chairman

1. "Friction and Oil Feed Characteristics of Journal Box Bearings."
George L. Pigman, Caterpillar Tractor Co., Peoria, Ill., and W. M. Keller, AAR
2. "Car Oil Additives."
Ray McBrian and L. C. Atchinson, Denver & Rio Grande Western R.R., Denver, Colo.
3. "Progress Report on Hot Boxes."
Victor C. Barth, Chicago and Northwestern R.R.
4. "Progress Report on Use of Molybdenum Disulfide in Railroad Journal Bearings."
E. E. Smith, Climax Molybdenum Co., New York, New York

**SESSION 5C Lubricating Mechanisms of Solids Lubricants**

Robert L. Johnson, National Advisory Committee on Aeronautics, Cleveland, Ohio, Organizing and Session Chairman

1. "Plastics, Solid Lubricants, and Bearings."
A. J. G. Allan, Polymer Chemicals Dept., E. I. du Pont de Nemours & Co., Inc.
2. "Molybdenum Disulfide and Related Solid Lubricants."
Bernard C. Stupp, Hohman Plating & Mfg. Co., Inc., Dayton, Ohio
3. "Mechanism of $WS_2$ Lubrication in Vacuum."
Melvin T. Lavik, Midwest Research Institute, Kansas City, Missouri
4. "Friction of Teflon as a Dry Lubricant Bearing."
S. B. Twiss, P. J. Wilson, and E. J. Snyder, Chrysler Corp., Engineering Div., Detroit, Michigan

**SESSION 5D Lubrication Engineering Course**

**SESSION 5E Exhibits**

**2:00-4:30 P.M.**

**SESSION 6B Lubrication Equipment**

E. M. Higgins, Master Lubricants Co., Organizing Chairman

T. M. Murphy, Stewart-Warner Corp., Chicago, Ill., Session Chairman

1. "Filtration of Coolants."
Gordon Signor, Scovill Mfg. Co., Waterbury, Conn.
2. "Filtration of Lubrication Oils."
Herbert A. Wilson, Commercial Filters Co., Melrose, Mass.
3. "Filtration of Diesel Engine Oils."
W. King Simpson, Electromotive Corp., La Grange, Illinois

**SESSION 6C Lubrication of Non-Ferrous Metals and Surfaces**

Dr. L. B. Sargent, Jr., Organizing and Session Chairman, Aluminum Co. of America, Pittsburgh, Pa.

1. "Wear and Frictional Characteristics of Some Wrought and Cast Nickel Base Alloys."
R. B. Kozlik, International Nickel Co.
2. "Mill and Laboratory Evaluation of Oils for Rolling of Copper Alloys."
F. L. Reynolds, Bridgeport Brass Co.
3. "Frictional Characteristics of Aircraft Wheel Brake Materials."
Dr. C. Sump, Armour Research Foundation, Chicago, Ill.

**SESSION 6D Reclamation and Maintenance**

N. C. Penfold, Southwest Research, San Antonio, Texas, Organizing Chairman

1. "Filtration of Motor Oil."
J. R. Griffin, Jr., Dev. Head, Lubrication Oil Petroleum Lab., E. I. du Pont de Nemours Co.
2. "Lubrication Problems Associated with Large Compressor Stationery Engines."
Clyde M. Floyd, Humble Oil & Refining Co., Houston, Texas

**SESSION 6D Exhibits**

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(New Products, from p. 7)

radius. Any combination of assembly lengths, fittings, and accessories are available, allowing the Kling-Flo to be used on any metalworking machine. (Cobon Plastics Corp., 44 Lafayette St., Newark 2, N. J.)

**Teflon Tape & Sheets.** New cementable Teflon tape is now available in continuous rolls in thicknesses from .005″ to .060″, in width from ½″ to 12″. Also available are cementable Teflon sheets 24″ x 24″ to 48″ x 48″, in thicknesses from 1/32″ to 1″. Nontoxic, non-contaminating, chemically-impervious Teflon tape is furnished with a slippery surface on one side and a specially prepared surface on the other side that can be bonded with commercial adhesives to wood, glass, metal, and other surfaces. Garlock 201 adhesive is recommended for cementing either the tape or sheet to common substrate materials such as wood, metals, glass, or plastics. (The Garlock Packing Co., Palmyra, N. Y.)

**Wheel Bearing Safety Service Kit.** A recently introduced Wheel Bearing Service Kit provides all the tools necessary to make the lubrication of front wheels a speedy, efficient, and profitable service for the car dealer, garage, or service station. Included in this new portable unit are tools for knocking out the union bearing, block for safe replacement of the seals, plastic-type hammer, pliers, container for new cotter pins, and also a tool for removing both the wheel cover and bearing dust cover. The Kit is light for easy handling, weighs approximately 20 lbs., with space in the cabinet for clean paper towels and a small size container for cleaning fluid. (The Aro Equipment Corp., Bryan, Ohio.)

## ASLE Preprints Available

A limited supply of the following preprints of papers presented at the ASLE 1956 Annual Meeting, and the ASLE-ASME 1956 Lubrication Conference, are still available at 35c each to members, 50c each to non-members. To order, indicate the number of copies desired, include your name and address, enclose remittance, and mail to:

**ASLE, 84 E. Randolph St., Chicago 1, Ill.**

_____(AM3C-1) *Analysis of Equilibrium Operating Temperatures of Railroad Journal Bearings*, by G. L. Pigman & W. M. Keller

_____(56LC-7) *Application of Reaction Rate Theory to the Weld Junction Aspect of Kinetic Friction Between Unlubricated Metallic Surfaces*, by F. F. Ling & E. Saible

_____(56LC-1) *Application Problems with Petroleum Lubricants in Nuclear Power Plants*, by R. F. Hausman & E. R. Booser

_____(AM4C-3) *Appraisal of Rolling Oil Testing, An*, by S. A. Paradee

_____(AM2C-2) *Design & Testing Considerations of Lubricants for Gear Applications*, by E. E. Shipley

_____(AM2B-2) *Dynamic Demulsibility Characteristics of Oils*, by E. W. Brennan & R. G. Moyer

_____(AM3B-2) *Effect of Engine Operation on Synthetic Gas Turbine Lubricants*, by J. H. Way & T. F. Davidson

_____(56LC-5) *Effects of Nuclear Radiation on Lubricants*, by J. A. King & W. L. R. Rice

_____(AM6A-2) *Electron Diffraction in Lubrication Research*, by D. Godfrey

_____(AM1B-2) *Evaluation of Anti-Scuff Properties of Oils*, by D. M. Teague, E. H. Loeser, P. J. Willson & S. B. Twiss

_____(AM1B-3) *Evaluation of Anti-Wear Properties of Oils*, by S. B. Twiss, E. H. Loeser & D. M. Teague

_____(AM3C-2) *Experimental Investigation of Railroad Journal Bearing Operating Characteristics*, by G. L. Pigman & A. L. Busby

_____(56LC-9) *Fretting of Hardened Steel in Oil*, by J. R. McDowell

_____(56LC-6) *Fundamental Study of Synthetic Sapphire as a Bearing Material, A*, by L. F. Coffin, Jr.

_____(56LC-2) *General Radiation Damage Problems for Lubricant & Bearing-Type Materials*, by V. P. Calkins & C. G. Collins

_____(AM3A-1) *Grease Flow in Shielded Bearings*, by R. O'Halloran, J. J. Kolfenbach & H. L. Leland

_____(56LC-18) *Influence of Temperature on Boundary Lubrication*, by C. W. Cowley, D. J. Ultee & C. W. West

_____(56LC-13) *Investigation of "Melt Lubrication,"* by B. Sternlicht & H. Apkarian

_____(AM3C-3) *Investigation of Oil Additives for Boundary Lubrication of Railroad Journal Bearings*, by A. Miller & A. A. Anderson

_____(AM2C-1) *Lubrication Engineer Surveys Steel Mill Gearing, A*, by A. E. Cichelli

_____(AM5A-1) *Lubrication of Aircraft Oscillating Control Bearings at High Temperatures*, by D. C. McGahey & R. S. Barnett

_____(AM4A-2) *Numerical Solution of Reynolds' Equation for Sector Thrust Bearings*, by B. Sternlicht & H. J. Sneck, Jr.

_____(56LC-10) *PbO & Other Metal Oxides as Solid Lubricants for Temperatures to 1000° F.*, by M. B. Peterson & R. L. Johnson

_____(AM4B-1) *Phosphate-Base Fluids*, by F. H. Langenfeld

_____(AM2A-3) *Problems in Hydrodynamic Lubrication*, by E. K. Gatcombe

_____(56LC-4) *Radiation-Resistant Greases*, by J. G. Carroll, R. O. Bolt & B. W. Hotten

_____(56LC-8) *Solid Film Lubricant—Factors Influencing Their Mechanism of Friction & Wear*, by R. E. Crump

_____(56LE-12) *Solution for the Finite Journal Bearing & Its Application to Analysis & Design — II, A*, by A. A. Raimondi & J. Boyd

_____(56LC-16) *Specific Heat at Low Temperatures and Latent Heat of Fusion of Aircraft Engine Lubricants*, by V. E. Shrock & R. E. Gott

_____(AM1A-2) *Spray Application of Lubricants to Plain Roll Neck Bearings*, by J. S. Aarons & C. M. Winn

_____(AM2A-1) *Surface Viscosity & Elasticity of Lubricating Oils*, by D. W. Criddle

_____(AM2B-3) *Synthetic Ester Lubricants with Improved Lubricity & Thermal Stability*, by M. Z. Fainman & R. S. Barnes

_____(56LC-11) *Temperature Effects in Journal Bearing Lubrication*, by W. F. Hughes & J. F. Osterle

_____(56LC-17) *Thermal Conductivity & Thermal Diffusivity of Aircraft Engine Lubricants at Low Temperatures*, by V. E. Schrock & E. S. Starkman

_____(56LC-3) *Use of Rolling Contact Bearings in Low Viscosity Liquid Metal Lubricants*, by W. Markert, Jr. & K. M. Ferguson

_____(AM6C-1) *Where We Stand on Fire-Resistant Turbine Fluids:A Progress Report*, by J. J. O'Connor

_____(56LC-14) *Yield Stress as a Factor in the Performance of Greases*, by D. Evans, J. F. Hutton & J. B. Matthews

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# Lubrication Summarized

**The Combustion-Chamber Deposit Problem in High-Compression-Ratio Engines,** by M. W. Corzilius, D. R. Diggs & R. A. Hoffman (E. I. duPont deNemours & Co., Inc.); presented at the 21st Midyear Meeting of the American Petroleum Institute's Division of Refining, May 15, 1956. (Summarized by N. C. Penfold, Southwest Research Institute.)

Perhaps the most important characteristic of a motor fuel which can be recognized by the motorist is that associated with its freedom from knock. The petroleum companies have found it very important to be able to supply fuels which will be relatively free of knock in the majority of motor cars; however, since higher antiknock fuels are more costly to produce, the refiner is continuously engaged in research to permit economic production of superior fuels. Automobile manufacturers have found it advantageous to increase engine compression ratios which, in turn, have caused a steadily-increased demand for knock-resistant fuels. Present indications are that compression ratios will reach 11:1 in the near future, and in all probability there will be incremental increases beyond that.

It has been known for some time that the engine, the fuel, and the lubricating oil all affect knocking tendencies, both because of inherent characteristics of the combustion chamber and deposits which are formed by the fuel and lubricating oil. The engine itself reacts differently to various fuels depending on the distributor, carburetor, and transmission characteristics. Further, pre-ignition or surface ignition is much more prevalent in engines as higher compression ratios are used, and fuels show significant differences in their resistance to surface ignition.

Realizing that the problem is a complex one involving not only the engine, fuel, and lubricating oil but also the type of driving, the authors undertook a program in the du Pont petroleum laboratory using extensive road tests in addition to the usual laboratory dynamometer tests. The road tests were made under controlled conditions, the cars being driven over a 350-mile course selected to provide the types of driving which might normally be encountered in passenger-car service including urban, suburban, country, and turnpike operation. Special attention was given to assessing the importance of engine design, compression ratio, fuel type, and oil type in relation to noise requirements. Noise requirements of the laboratory dynamometer-connected engines were determined after simulated road operation using the same reference fuel systems as the cars operated on the road. Engines with compression ratios as high as 11:1 were used in the study to determine the effect of compression ratio on equilibrium noise requirements. Three lubricating oils were used, one of which was a low-viscosity synthetic lubricant of the diester type, in order to determine the lubricant contribution to deposit harm. These studies were made in both the road test course and in the laboratory engines. It was found that multigrade lubricants were generally helpful in lowering deposit harm, and that the synthetic lubricant was substantially better, indicating that considerable improvement in lubricating oils can still be made.

In order to study the effect of different fuels on deposit harm, three fuel blends were prepared by adding high-boiling material to a base fuel which had been used in much of the deposit accumulation work. The materials were narrow-cut aromatics, heavy catalytic reformate, and heavy catalytic naphtha to which TEL was added to raise the level to 3.0 ml per gallon. The 90 per cent distilled point was not found to be a significant criterion for judging the deposit harm of a fuel.

The authors demonstrated that even with an 11:1 compression ratio an engine could be operated under normal conditions free of combustion noise on 100-octane number fuel by giving proper attention to engine design, fuel, and lubricating oil.

# Book Reviews

**"National Standards in a Modern Economy;"** Harper & Bros. publishers, 1956; available from American Standards Association, Dept. P-1, 70 E. 45th St., New York City; 372 pages, cloth bound, price $5.00.

"National Standards in a Modern Economy" is a definitive review of the evolution of industrial standards and their crucial place in the world today, by 34 leaders in the private and public organizations that have developed modern standards. (Among the authors are former president, Herbert Hoover; Howard Coonley, formerly president and chairman of the board of the Walworth Company; and Thomas D. Jolly, vice-president of the Aluminum Company of America.)

These authorities link standards with the achievements of scientific research and technological innovation that bring about high production efficiency in the age of automation. They discuss the vital role of standards in a variety of fields, such as: interchangeable parts manufacture, international relations, quality control, management, and research and development.

The volume was edited by the late Dickson Reck of the School of Business, University of California, and is a joint undertaking of the Standardization Fellow-

(Continued on p. 63)

# Lubrication Abstracts

(Compiled by the ASLE Abstracts Sub-Committees.)

## BEARINGS

**Very Short Journal-Bearing Hydrodynamic Performance Under Conditions Approaching Marginal Lubrication,** by L. F. Kreisle, ASME "Trans.," Vol. 78, No. 5, July 1956, pp. 955-961, discussion pp. 961-963. Experimental data on the hydrodynamic performance of full journal bearings (1⅛ inch dia.) under conditions approaching zero minimum oil film thickness are presented for bearings with length to diameter ratios between ½ and 1/53. Measurements of eccentricity, minimum film thickness, attitude angle, bearing friction, and oil flow are made. The experimental information is compared with several available theories and correlated through the use of dimensionless parameters which the author defines as load numbers. An empirical constant is found to improve the correlation. The author concludes that the presence of marginal lubrication is indicated when the minimum oil film thickness is less than the mean square surface roughness of the journal and bearing. (Abstracter: A. A. Raimondi)

**Tapered-Land Thrust Bearing,** by C. F. Kettleborough, ASME Paper No. 56-APM-21, presented June 14-16, 1956. A composite slider consisting of a plane surface tapered in the direction of motion, followed by a flat surface, is studied analytically using relaxation techniques. Numerical calculations are made for square pads, and account for side leakage; constant viscosity is assumed. It is pointed out that this tapered-land bearing does not suffer from a certain disadvantage of the stepped slider bearing, and is capable (over a limited range of operating conditions) of supporting about 14% greater load than the maximum load carried by a pivoted-pad bearing. (Abstracter: A. A. Raimondi)

**Over-Lubrication Shortens Anti-Friction Bearing Life,** by E. P. Stahl, "Iron Age," Vol. 178, No. 12, September 20, 1956, pp. 91-93. Over-lubrication is a common cause of bearing failure; its prevention by use of pressure-relief fittings on bearing housings is described. When to use oil and when to use grease for bearings, the qualities required of lubricants, the avoidance of over- and under-lubrication, the types of seals, and the time interval for replenishing the lubricant are discussed. (Abstracter: A. A. Raimondi)

**The Sector-Shaped Pad,** by C. F. Kettleborough, ASME Paper No. 56-APM-22, presented June 14-16, 1956. Boegli's simplifying assumptions for the solution of the rectangular pad have been applied to the sector-shaped pad and extended to the case where the viscosity varies linearly. The process of obtaining operating characteristics such as load, friction, and center of pressure, is rapid and yields results comparable with those obtained by longer numerical methods. (Abstracter: A. A. Raimondi)

**Journal-Bearing Operation at Superlaminar Speeds,** by M. I. Smith & D. D. Fuller, ASME "Trans.," Vol. 78, No. 3, April 1956, pp. 469-474. Investigations of load capacity and friction characteristics on a 3" x 3" water-lubricated full journal bearing are reported for speeds ranging up to five times the critical speed of transition from laminar to turbulent flow. It is found that the frictional behavior of an unloaded bearing is such as to allow definition of regions described as laminar, transition, and turbulent. The critical speed for an unloaded bearing is in good agreement with that predicted by Taylor's instability criterion. With increasing penetration into the turbulent region, the bearing exhibits greater load-carrying capacity and friction than would be calculated by the laminar solution of Cameron and Wood. Some shaft whirl is experienced. (Abstracter: A. A. Raimondi)

## LUBRICANTS

**Friction & Wear Studies of Chlorinated Methylphenyl Silicones,** by R. C. Bowers, R. L. Cottington, T. M. Thomas & W. A. Zisman, "Indus. & Engr. Chem.," Vol. 48, No. 5, May 1956, pp. 943-950. Stick-slip apparatus and 4-ball wear machines are used to evaluate the lubricity of four chlorine-containing silicones. Combinations steel-steel and steel-copper are used from 77 to 600° F.; lubricity is usually improved as the chlorine content is increased. Metal-liquid interfacial reaction appears responsible for improvement over conventional silicones. Machinery must be designed to prevent boundary conditions of operation when silicones are used with steel on steel. (Abstracter: C. C. Currie)


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# Patent Abstracts

(Compiled by Ann Burchick, Aluminum Co. of America.)

**Lubricant Compositions and Method of Making Same,** Patent No. 2,733,210 (E. R. Taylor, Jr., assignor to E. I. duPont deNemours & Co.) A homogeneous single phase liquid lubricant composition consisting essentially of (1) an aqueous solution of a 1-4 carbon aliphatic monohydric alcohol in which the ratio of alcohol to water is between about 45:55 and about 10:30, (2) 5.0 to 30.0%, based on total weight, of a water-insoluble monohydroxy 1,2-polyoxypropylene monoalkyl ether having an average of at least nine oxy 1,2-propylene groups per molecule and up to 20 carbon atoms in the alkyl group, and (3) 0.1% to 0.80%, based on total weight, of a dibasic borate salt of an oxygen containing amine selected from the group consisting of monoethanol amine, 2 amino 2 methyl propanol-1, morpholine, N-methyl morpholine, N-ethyl morpholine, isopropyl amino ethanol, N-methyl ethanol amine, N-dimethyl ethanol amine, N-ethyl ethanol amine, N-diethyl ethanol amine and isopropanol amine.

**Mineral Oil Additive,** Patent No. 2,733,235 (T. Cross, Jr. & J. H. Bartlett, assignors to Esso Research & Engrg. Co.) As a new composition of matter a reaction product selected from the class consisting of: (A) the reaction products obtained by reacting, at a temperature in the range of about 200 to about 500 F. a reactive sulfur and phosphorus-containing material with an ester of an alkenyl succinic acid.

**Mixed Soap-Complex Grease Composition,** Patent No. 2,734,030 (E. O. Forster & J. P. Hamer, assignors to Esso Research & Engrg. Co.) A lubricating grease composition having outstanding high temperature properties which comprises a lubricating oil thickened with from about 5.0 to 30.0% by weight, based on the weight of the total composition, of a complex formed from a mixture of a calcium soap and a second soap selected from the group consisting of barium, strontium and magnesium soaps of high molecular weight fatty acids and a metal salt of a low molecular weight carboxylic acid having from one to six carbon atoms, said metal being selected from the group consisting of calcium, barium, magnesium and strontium, the calcium soap being present in an amount ranging from about 2.5 to 15.0%, by weight, based on the weight of the total composition, the mol ratio of said calcium soap to said second soap being about 8:1 to 1:8, the mol ratio of said mixture of soaps to said metal salt being within the range of about 0.70 to 1.2, said complex being formed by first dispersing the calcium soap in the lubricating oil, then dispersing the second soap and said metal salt, and heating the resulting mixture to a transition temperature from about 350 to 480 F.

**Lubricants Containing Polymers of Cyclopentadiene,** Patent No. 2,734,031 (T. J. McNaughtan, assignor, by mesne assignments to Hooker Electrochemical Co.) A lubricating composition comprising an organic lubricating liquid of oily nature and a polymer of cyclopentadiene having the empirical formula of $(C_5H_6)_n$ in which n is a figure of at least 3 and wherein there are two double bonds, the per cent by weight of polymer present based on the weight of the organic lubricating liquid of oily nature in the composition being sufficient to increase and stabilize the viscosity of the organic liquid but not more than 25% by weight thereof.

**Lubricants,** Patent No. 2,734,032 (W. J. Coppock, assignor to Sun Oil Co.) A lubricating composition comprising a major proportion of mineral lubricating oil as the essential lubricating component, a minor proportion of an alkylated phenol oxidation inhibitor as antioxidant.

**Biguanide Derivatives as Corrosion Inhibitors,** Patent No. 2,734,807 (J. A. Chenicek & R. B. Thompson, assignors to Universal Oil Products Co.) In the transportation and storage of hydrocarbon oils where the oil contacts metal surfaces in the presence of water, the method of inhibiting corrosion of the metal surfaces by said water which comprises dissolving in the hydrocarbon oil from about 0.0001% to about 2% by weight of a substituted biguanide having a hydrocarbon substituent on at least one of its terminal nitrogen atoms.

**Synthetic Casein Fiber Elastic Grease,** Patent No. 2,734,862 (A. J. Morway & R. P. Spray, assignors to Esso Research & Engrg. Co.) An elastic fiber lubricant composition comprising a major proportion by weight of a lubricating grease and a minor proportion by weight of a fibrous mass comprising a major proportion of a mildly elastic wool fiber and about 10 to 40% by weight of a highly elastic extruded monofilament curled casein fiber, which has strong resistance against matting at temperatures up to at least 200 F.

**Phospho-Sulfurized Hydrocarbons,** Patent No. 2,734,863 (R. H. Jones, M. W. Hill & L. E. Moody, assignors to Esso Research & Engrg. Co.) A hydrocarbon oil composition containing a detergent quantity of a product selected from the class consisting of a phosphosulfurized substantially non-olefinic hydrocarbon materials and the guanidino and reactive olefinic derivatives thereof, said materials containing at least 1.0 weight per cent of both phosphorus and sulfur in chemically combined form; which product has been contacted during at least one stage of the preparation thereof with a metal selected from the class consisting of iron and copper at elevated temperatures and for an extended period of time, adequate to produce a substantially odorless, non-corrosive product.

**Stabilized Lubricating Compositions,** Patent No. 2,734,864 (J. B. Peeso & E. O. Hook, assignors to American Cyanamid Co.) A hydrocarbon mineral lubricating oil composition comprising a hydrocarbon mineral lubricating oil and, as an antioxidant and anticorrosive agent having a tendency toward heat instability and the formation of sludge upon continued heating thereof at high temperatures, 0.1-3.0% by weight of a dialkyl dithiophosphate.

**Stabilized Lubricating Compositions,** Patent No. 2,734, 865 (J. B. Peeso & E. O. Hook, assignors to American Cyanamid Co.) A hydrocarbon mineral lubricating oil composition comprising a hydrocarbon mineral lubricating oil and, as an antioxidant and anticorrosive agent having a tendency toward heat instability and the formation of sludge upon continued heating thereof at high temperatures, 0.1-3.0% by weight of a dithiophosphate.

**Stabilized Lubricating Composition,** Patent No. 2,734,866 (E. O. Hook & J. B. Peeso, assignors to American Cyanamid Co.) A hydrocarbon mineral lubricating oil composition comprising hydrocarbon mineral lubricating oil and, as an anti-oxidant and anti-corrosive agent having a tendency toward heat instability and the formation of sludge upon continued heating thereof at high temperatures, 0.2-3.0% by weight of a terpene-phosphorus sulfide-sulfur reaction product and, as a heat stabilizer, 0.3-1.0% by weight of a neutralized phosphorus sulfide-oxygen reaction product with a member of the group consisting of esters of alcohols and tall oil faty acids and mixtures of alcohols and tall oil fatty acids.

**Additive for Lubricants,** Patent No. 2,734,867 (J. S. Weissberg & J. R. Jones, assignors to Esso Research & Engrg. Co.) A lubricant composition comprising a major proportion of a lubricating oil and in the range of 0.01 to 10% by weight of the product defined by claim 9.

**Rock Drill Lubricant Containing a Phosphorus acid Ester and a Sulfurized Fatty Oil,** Patent No. 2,734,868 (G. S. Bright & R. W. Hall, assignors to The Texas Co.) A pneumatic tool lubricant

consisting essentially of a mineral lubricating oil having a viscosity in the range of from about 300 to 700 seconds Saybolt Universal at 100 F., containing about 0.2 to 3.0 per cent by weight of a phosphorus acid ester selected from the class of oil-soluble and oil-miscible di- and tri- alkyl, aryl and mixed alkyl-aryl phosphates and phosphites, about 0.2 to 5.0 per cent by weight of a non-corrosive sulfurized fatty oil containing at least about 4 per cent by weight of combined sulfur, about 0.1 to 1.0 per cent by weight of a phosphatide emulsifying agent selected from the class consisting of lecithin, cephalin and sphengomyelin, and about 0.1 to 3.0 per cent by weight of a high molecular weight polymeric stringiness agent selected from the class consisting of olefin polymers vinyl ether polymers and polyesters of substituted fatty acids, said stringiness agent having a Saybolt furol viscosity at 210 F. of about 1000 to 2000 seconds.

**Lubricating Oil Additive,** Patent No. 2,734,869 (L. V. Mullen, Jr. & J. M. Boyle, assignors to Esso Research & Engrg. Co.) A composition comprising a major proportion of a petroleum hydrocarbon product and a minor oxidation inhibiting quantity of product obtained by reacting a mono-carbonyl ketone containing at least one olefinic double bond, and having in the range of 4 to 30 carbon atoms, with a sulfur halide selected from the class consisting of sulfur chlorides and sulfur bromides.

**Die Forging Compound,** Patent No. 2,735,814 (L. N. Hodson, Sr. & T. C. Foin, assignors to The Hodson Corp.) A die forging composition comprising, on a weight basis, 5 to 30% graphite, 2 to 15% hemicellulose, 1 to 15% of kettle bodied fish oil, 2.5 to 6% bentonite, 0.15 to 2.5% polyethylene glycol monooleate, and the balance water to 100%.

**Stabilized Lubricating Oil Additives,** Patent No. 2,735,817 (W. E. Waddey, M. W. Hill & F. B. Cyphers, assignors to Esso Research & Engrg. Co.) A process for reducing the tendency of a phosphorus sulfide treated sulfurized sperm oil to evolve hydrogen sulfide on storage which comprises treating said phosphorus sulfide treated sulfurized sperm oil with about 1 to 5% of its weight of guanidine carbonate in aqueous solution at a temperature between about 210 to 300 F. for a time sufficient to reduce its tendency to evolve $H_2S$ on storage and to evaporate essentially all of the water, and removing normally solid undissolved constituents therefrom.

**Fuel Oil Additive,** Patent No. 2,736,641 (R. W. Mattson & F. S. Scott, assignors to Union Oil Co. of Calif.) A fuel oil containing a minor proportion, sufficient to inhibit the separation of sludge, of a surface active amido alcohol consisting essentially of a fatty acid amide of a lower alkanolamine, said fatty acid containing from 10 to 18 carbon atoms.

**Cold Working Lubricant for Wire and the Like,** Patent No. 2,736,699 (R. L. Carhart, assignor to United States Steel Corp.) A cold working lubricant for wire or the like comprising a mixture of from 8 to 35% water insoluble metal soap, from 20 to 80% of a substance from the group consisting of ferrous sulphate, ferric sulphate and a combination thereof, each molecule of said ferrous sulphate and each molecule of said ferric sulphate having at least four molecules of water of crystallization, and from 10 to 45% of a filler compound having a metal radical which is above ferric iron in the electromotive series and which will satisfy the sulphate radical of said ferrous sulphate and the sulphate radical of said ferric sulphate, said filler being substantially inert with respect to the wire and being a compound from the group consisting of the borates, silicates, phosphates, carbonates, chromates, hydroxides, oxides and sulphides of metals which are above ferric iron in the electromotive series, the percentages being by weight.

**Lubricant for Cold Drawing of Thorium Wire,** Patent No. 2,736,700 (C. D. Graham, Jr. & R. J. Donley, assignors to the United States of America as represented by the U. S. Atomic Energy Commission) A wire drawing composition consisting essentially of a mixture of 2 to 5 parts by weight molybdenum disulfide and 1 part by weight of finely divided leaded zinc oxide suspended in a drying vehicle.

**Lubricating Oil Composition,** Patent No. 2,736,701 (L. L. Neff, assignor to Union Oil Co. of Calif.) A lubricating oil additive concentrate adapted for addition to mineral lubricating oil which comprises a mineral oil solution containing 15% to 50% of an oil soluble complex obtained by heating an oil solution of an oil soluble metal salt of a hydrocarbon substituted phenol-aldehyde resin, in which the aldehyde has not more than 5 carbon atoms per molecule, to a temperature between about 325 F. and about 400 F., adding to the heated solution an aqueous mixture of at least one basically reacting metal inorganic compound selected from the class consisting of the hydroxides, carbonates and bicarbonates of barium, strontium, sodium, potassium and lithium, said mixture containing between about 25% and about 90% by weight of said compound and being added at a rate such that the addition takes place over a period of at least about 30 minutes and continuing the heating for a time sufficient to volatilize water thereby solubilizing between about 0.1 and about 3 equivalents of said basically reacting metal inorganic compound per equivalent of phenol-aldehyde resin salt in the oil solution of said resin salt.

**Detergent Lubricating Oils,** Patent No. 2,736,704 (T. L. Cantrell & H. G. Smith, assignors to Gulf Oil Corp.) A lubricant composition comprising a major amount of a mineral lubricating oil and a minor amount, sufficient to confer detergency properties on the composition, of an alkaline earth metal salt of a condensation product prepared by the process which comprises reacting a mixture of about 6 mols of a monoalkylated monohydric phenol, about 2 mols of aniline and about 1 mol of carbon disulfide, cooling the resulting mixture to room temperature, and then adding said cooled mixture to an aqueous slurry of about 3 mols of an alkaline earth metal hydroxide and about 8 mols of formaldehyde, and raising the temperature sufficiently to distill off all water.

**Detergent Lubricating Oils,** Patent No. 2,736,705 (T. L. Cantrell & H. G. Smith, assignors to Gulf Oil Corp.) A lubricant composition comprising a major amount of a mineral lubricating oil and a minor amount, sufficient to confer detergency properties on the composition, of a product prepared by dehydrating the product of metathesis of: (1) a condensation product prepared by the process which comprises reacting a mixture of about 4 mols of a monohydric monoalkylated phenol and about 2 mols of an alkali metal hydroxide in aqueous solution, cooling the resulting mixture, adding thereto about 2 mols of carbon disulfide and additional water, heating to about 200 F., then adding thereto a mixture of about 1 mol of an alkaline earth metal hydroxide in an aqueous slurry and about 4 mols of formaldehyde and reacting the resulting mixture; with (2) an aqueous solution of a water-insoluble salt of an alkaline earth metal.

**Treatment of Lubricating Oil with Adsorbent Clay,** (Patent No. 2,738,314 C. J. Lewis & J. R. Miller, assignors, by mesne assignments, to Esso Research & Engrg. Co.) A method for improving the color of a lubricating oil fraction which comprises contacting the lubricating oil fraction with an adsorbent clay at a temperature within the range of 300 to 500 F. in the presence of a mixture of approximately equal amounts within the range of .14% to .28% of alkali metal salts of naphthenic and sulfonic acids.

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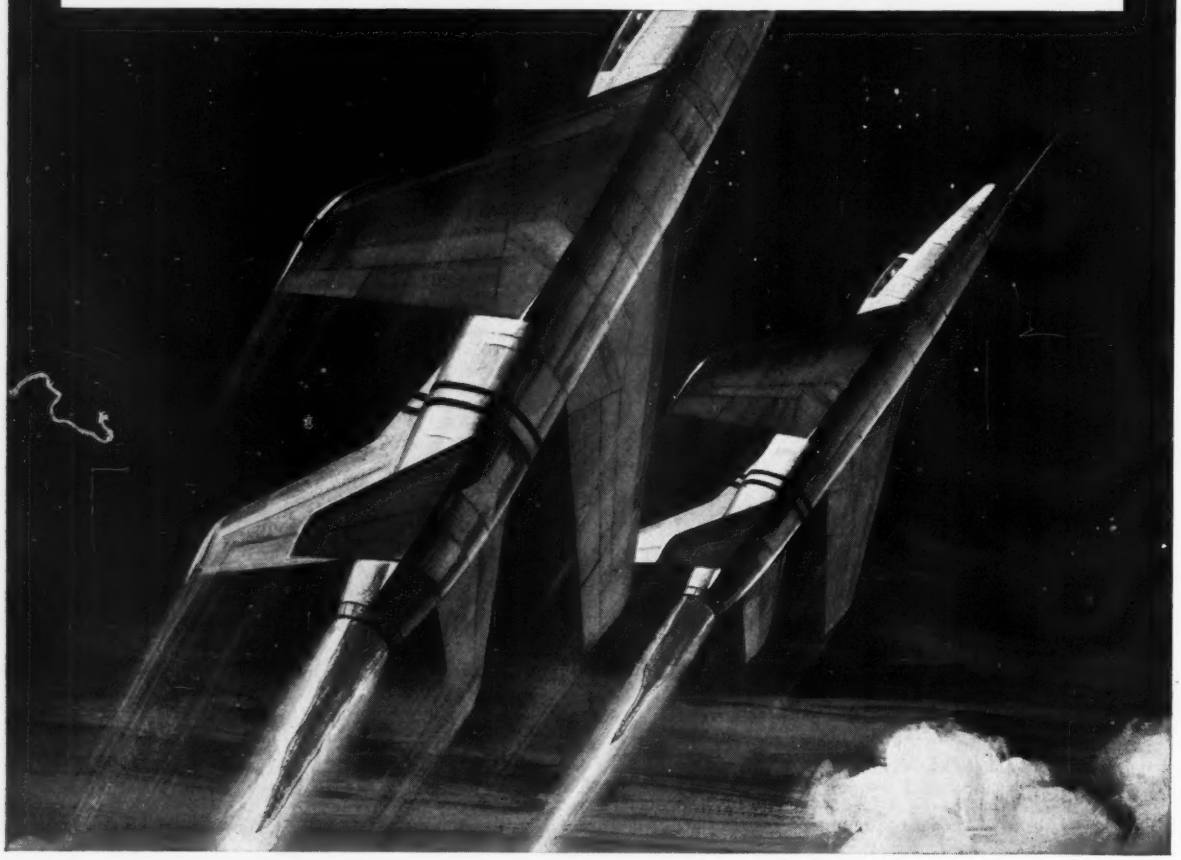

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(Men of Lubrication, from p. 13)
carrying an extra gun or ammunition. Tower then tried to adapt his ideas to a scheme to provide "steady seats" on cross-Channel steamers. He was engaged in this work at the time of his death.

Tower married in 1902. He died suddenly from a cerebral hemorrhage at his home in Bretwood Essex in 1904.

ACKNOWLEDGEMENTS. The principal sources of information consulted in the preparation of this article were obituary notices in the transactions of professional societies, and a short account of Tower's career in "Life of Lord Rayleigh" by the Fourth Baron Rayleigh. The author is also indebted to Miss Cecily Tower and to Lord Dunboyne —distant relatives of Beauchamp Tower—and, in particular, to Major R. B. Tower, his nephew, who provided the very fine picture here reproduced.

(This is the ninth in a series of articles by F. R. Archibald, Analyst, of Arthur D. Little, Inc.)

(Book Reviews, from p. 58)
ship at Mellon Institute of Industrial Research in Pittsburgh, and the American Standards Association.

(Foreign Abstracts, from p. 16)
tween fundamental wear behavior of steel and that of copper and zinc. Data on influence of speed of grinding wheel used for tests on amount of stock removed; comparison of these data with results obtained at slow wheel speeds and with artificial heating of specimens. Interpretation of results. (Order No. 3062, price $2.90)

**Study of Antifriction Properties of Chromium Deposits Made Porous by Mechanical Means,** by D. N. Garkunov & A. A. Polyakov; "Vestnik Mashinostroeniya," Vol. 33, No. 6, 1953, pp. 65-67; 5 figs., 1400 words. Study of frictional wear and friction coefficient of gray iron (of the type used for piston rings) against conventional porous chromium deposits and chromium deposits rendered porous by mechanical means. Wear testing machine (Fig. 1) and procedure used by authors. Data on variation of friction coefficient (Fig. 4) and frictional wear (weight loss) as function of specific load and amount of lubricant used, with the two types of chromium deposits tested. Behavior of these deposits in the absence of any lubrication (Fig. 5). (Order No. 3134, price $3.60)

**Effect of Lubricants upon Coefficient of Friction in the Deep Drawing of Metals,** by S. Ya Veiler & G. I. Epifanov; "Doklady Akademii Nauk SSSR," Vol. 92, No. 3, 1953, pp. 593-595; 1 fig., 1200 words. Study of part played by external friction (between work piece and tool) in the deep drawing of metals (0.05-0.15% plain carbon steel; brass, and copper). Experimental procedure. Data on coefficient of friction as function of number of draws (dry, methyl alcohol, butyl alcohol, and nonanoic acid) for steel, brass, and copper. Conclusions drawn from results of study: significance of plasticizing action of lubricants. (Order No. 3248, price $2.40)

**Formation & Breakdown of Welding Bridges Between Friction Surfaces,** by M. P. Levitskii; "Doklady Akademii Nauk SSSR," Vol. 92, No. 4, 1953, pp. 797-798; 1 fig., 750 words. Processes of formation and breakdown of welding bridges between rubbing surfaces studied in the light of the kinetic theory of process rates and the active-complex theory. Author's contribution to the concept that part of the frictional force is due to the formation of welding bridges at true contact points between rubbing surfaces. Formulae developed in support of the modified concept. (Order No. 3403, price $1.90)

**Wear Resistance of Steel Surfaces.** by V. V. Chernyshev; "Vestnik Mashinostroeniya," Vol. 32, No. 9, 1952, pp. 54-57; 6 figs., 3 tables, 1900 words. Study of wear of various steels at different sliding speeds and specific pressures, between two cylindrical bodies, one stationary, the other rotating. Experimental setup (Fig. 2). Content of alloying elements in low alloy steels (up to 1.65% Mn, 0.3% Si, 1% Cr, 0.5 C, and 0.4% Ni) as affecting the wear resistance. Data on 0.45% C steel. Comparative effects of surface hardening as against full hardening. Effect of tempering temperature. Conclusions drawn from results. (Order No. 3450, price $3.80)

**Wear Caused by Metal-Against-Metal Sliding Friction, with Special Reference to the Effect of Temperature,** (Part I), by W. Radeker; "Archiv f.d. Eisenhüttenwesen," Vol. 15, No. 10, 1942, pp. 453-46); 10 figs., 5 tables, 5700 words. Critical review of literature. Study of

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process of wear by pure sliding friction at temperatures ranging from −310°F. to 1300°F. Materials studied: plain carbon steel with 0.04 to 0.73% C; 17% chromium steels with 0.1-0.9% C; 18 Cr, 8 Ni steel; austenitic manganese steel; gray cast iron; carbonyl nickel; etc. Experimental setup (permitting cooling and heating of specimens); procedure in testing; testing speeds: 6 and 30 ft/sec. The three different types of wear observed. Influence of travel, load, microstructure, and hardness upon wear. Wear as function of speed of sliding; characteristics of wear in water, air, machine oil, soluble oil, aq. soap solns., colloidal graphite in water; dry emery, dry oxide scale; emery in water; oxide scale in water; liquid air; 3% aq. hydrogen peroxide; fuel alcohol; gasoline; kerosene; 25% aq. sodium chloride; 10% aq. sugar soln.; emulsion of 4% soluble oil in water. (Order No. 3460, price $9.85)

(Part II, conclusion), pp. 460-469; 18 figs., 5200 words. Influence of temperature on the different types of wear of plain carbon, stainless Cr and Cr-Ni and austenitic Mn steels; also cast iron. Relationship between temperature and process of wear as a function of load and sliding velocity. Observations on changes of surfaces being subjected to wear; tongue and groove formation; flowed-metal layer; oxidic abrasion. Explanation of some peculiarities of wear under the conditions of experiment—wear at high vs. low sliding speeds; wear in water; wear at elevated external temperature; inconsistent effect of load on degree of wear. General regularities of metal-against-metal friction uncovered or confirmed, respectively, by author's study. (Order No. 3461, price $9.65)

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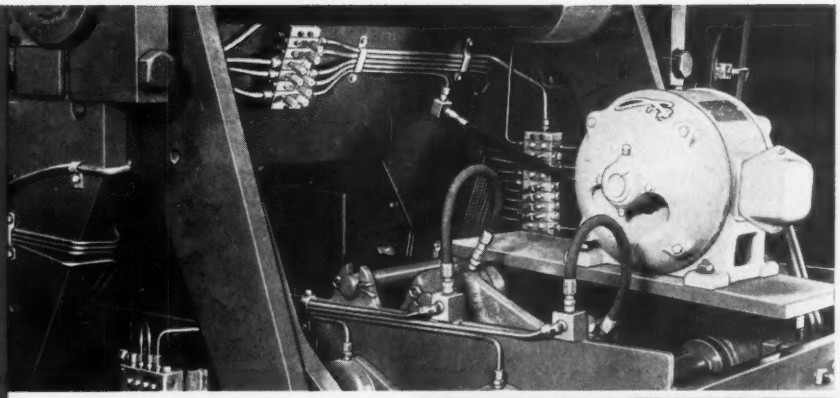

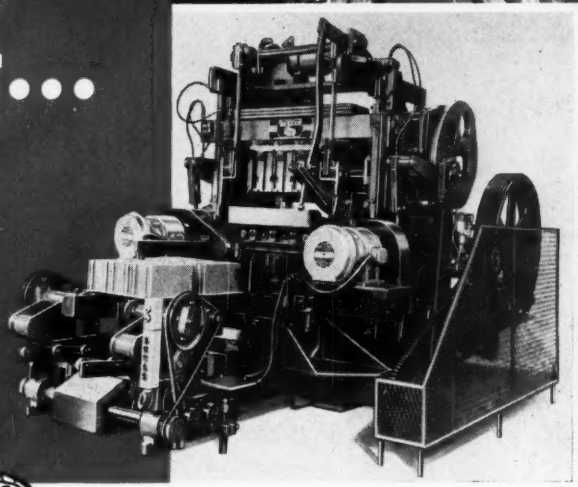

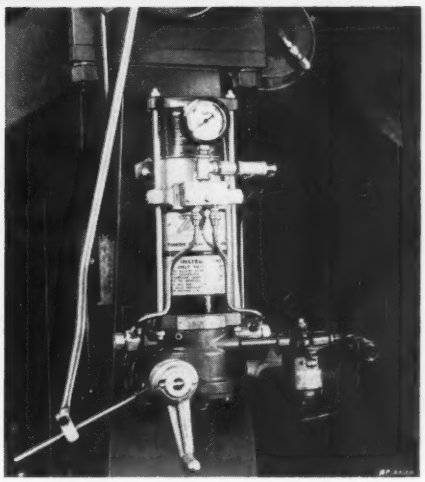

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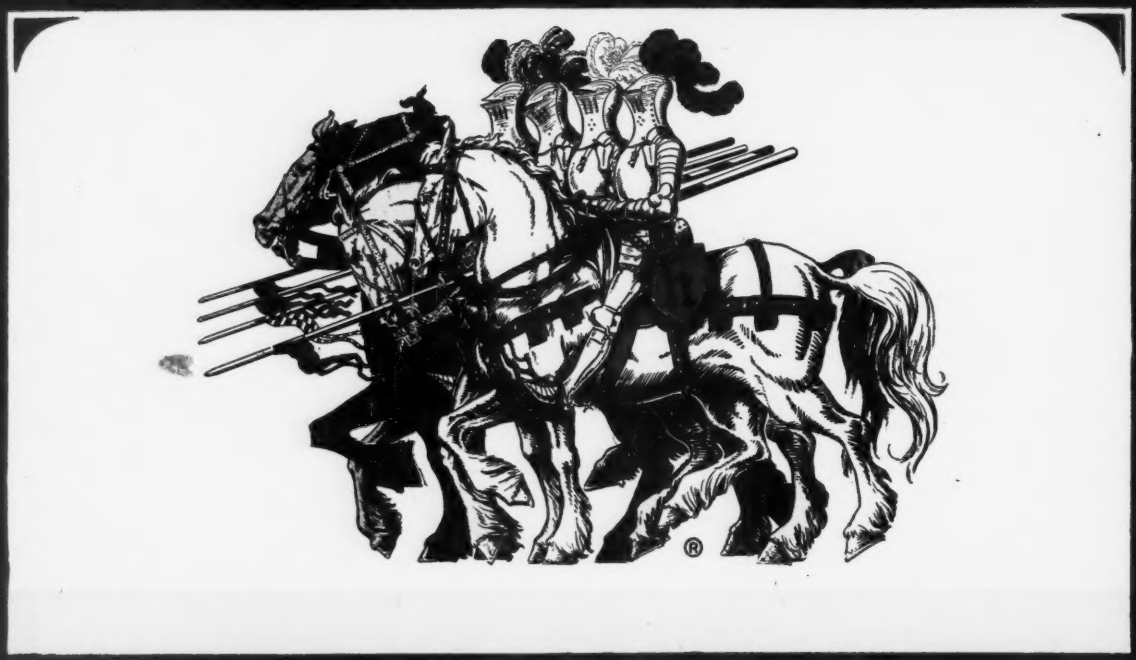

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